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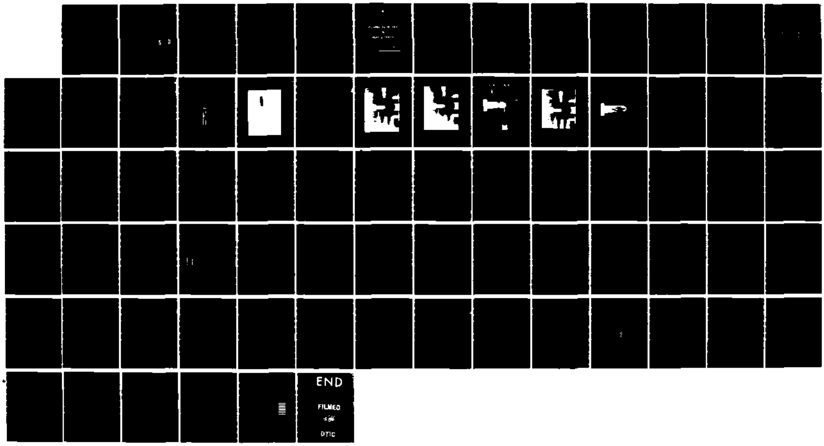

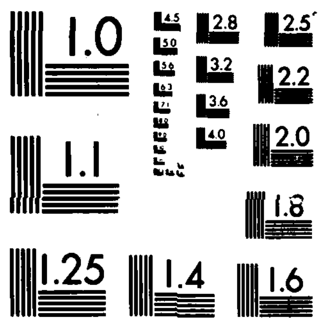


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# TECHNICAL NOTES ON ORIFICE TESTS

General Electric
Ordnance Systems Division
100 Plastics Avenue
Pittsfield, Massachusetts 01201



January 1986



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# REPORT DOCUMENTATION PAGE

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| | | |
|---|---|---|
| 1. REPORT NUMBER<br>Contract Report BRL-CR-548 | 2. GOVT ACCESSION NO.<br>AD-A164227 | 3. RECIPIENT'S CATALOG NUMBER |
| 4. TITLE (and Subtitle)<br>Technical Notes on Orifice Tests | | 5. TYPE OF REPORT & PERIOD COVERED<br>FINAL |
| | | 6. PERFORMING ORG. REPORT NUMBER |
| 7. AUTHOR(s)<br>Mel J. Bulman and John Mandzy | | 8. CONTRACT OR GRANT NUMBER(s)<br>DAAK 11-78-C-0054 |
| 9. PERFORMING ORGANIZATION NAME AND ADDRESS<br>General Electric Ordnance Systems Division<br>100 Plastics Avenue<br>Pittsfield, MA 01201 | | 10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS |
| 11. CONTROLLING OFFICE NAME AND ADDRESS<br>US AMC/Ballistic Research Laboratory<br>ATTN: SLCBR-DD-T<br>Aberdeen Proving Ground, MD 21005-5066 | | 12. REPORT DATE<br>January 1986<br>13. NUMBER OF PAGES<br>69 |
| 14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office) | | 15. SECURITY CLASS. (of this report)<br>UNCLASSIFIED<br>15a. DECLASSIFICATION/DOWNGRADING SCHEDULE |

16. DISTRIBUTION STATEMENT (of this Report)

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17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

high speed flow of propellant
flow visualization
ball check injector
straight hole injector
high pressure flow test fixture
flow cavitation
flow separation
augmented propellant combustion

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

The purpose of this task was to study the high speed flow of monopropellant through orifices in order to develop criteria for safe and efficient designs. Flow tests were conducted in a clear plastic, low pressure fixture and firing tests in a specially designed steel fixture. The initial injector studied was known to behave poorly in regenerative gun fixture firings. The flow tests revealed that it was subject to flow separation.

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20. ABSTRACT (Con't)

The steel fixture tests with this injector exhibited strong evidence for flameholding inside the injector. Steel fixture firing tests were also performed on straight-through hole injectors. These also showed evidence of flameholding in the orifice. The basic goal of this effort was not met. It is believed that neither the basic designs of the steel text fixture nor the instrumentation used were adequate for this task. Although no quantitative data was generated, a number of useful qualitative design guidelines were obtained.

GENERAL ELECTRIC

# TECHNICAL NOTES ON ORIFICE TESTS

CONTRACT DAAK 11-78-C-0054

ORDNANCE SYSTEMS DIVISION
100 PLASTICS AVENUE • PITTSFIELD, MASSACHUSETTS 01201

TECHNICAL NOTES ON ORIFICE TESTS†

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September, 1984



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†Work performed under Army Contract DAAK 11-78-C-0054

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## PREFACE

This series of Technical Notes was compiled in order to provide a reasonable level of access to the voluminous data generated under the ACT Program (Contract DAAK 11-78-C-0054). The intent was not to provide a finished product, well organized and thoroughly analyzed, but rather to provide a ready reference document. These documents are written around the raw data. Only sufficient organization was provided to allow access of the desired data without an undue amount of search time. In keeping with this spirit, speculations are presented on an equal footing with conclusions based on analyses. It is the hope of the authors that these notes will prove of benefit to future researchers in this field.

Section 1

INTRODUCTION

## 1.1 ORGANIZATION

These Notes are organized in the following manner. The purpose is presented in the Introduction, together with a brief description of the two fixtures used in this effort, the clear plastic flow tester and the steel orifice tester. Section 2 presents a description of the work performed using the plastic fixture together with an analysis and interpretation of the data from these tests. Section 3 presents the equivalent work using the steel orifice tester. A discussion of the results, and the conclusions reached therefrom, is presented in Section 4.

The possibility of flameholding occurring in the injector, and mechanisms for enhancing the propellant burning rate while it is in the orifice, are particularly germane to this investigation. A brief discussion of these two topics is presented in Appendices A and B, respectively.

## 1.2 BACKGROUND

These Notes were compiled to provide further background to the orifice flow investigation. The initial stages of this effort were reported in the "Liquid Propellant Technology Annual Report," February, 1980.[1] A detailed discussion of the purpose, the test fixtures used, their design principles, and the initial results, may be found in the above referenced document. For the purpose of continuity, a brief summary of this information is presented below.

The purpose of this task was to study the high speed flow of monopropellant through orifices in order to develop criteria for safe and efficient designs. A twofold approach to this problem was taken. First, the flow would be visualized in a small, low pressure fixture of clear plastic, using water as the working fluid. It was realized that the results obtained from this fixture could not be treated as more than qualitative guides. The difference in working pressure between the plastic fixture and a typical steel firing fixture is more than two orders of magnitude. A sketch of the plastic fixture, including the location of the pressure taps, is shown in Figure 1-1.

The plastic fixture was complemented, for the second part of the approach, by a high pressure steel fixture which would inject actual propellant into hot combustion chamber gases. An assembly drawing of the steel orifice tester is shown in Figure 1-2. The design goal for this fixture was to instrument the propellant flow passage (injector orifice) while still duplicating the environment of a regenerative liquid propellant gun. The approach adopted was to fix the injector orifice with respect to the chamber, which allowed instrumenting the flow passage to be relatively straightforward.

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[1] J. Mandzy, "Liquid Propellant Technology Annual Report," General Electric Ordnance Systems, 1980.

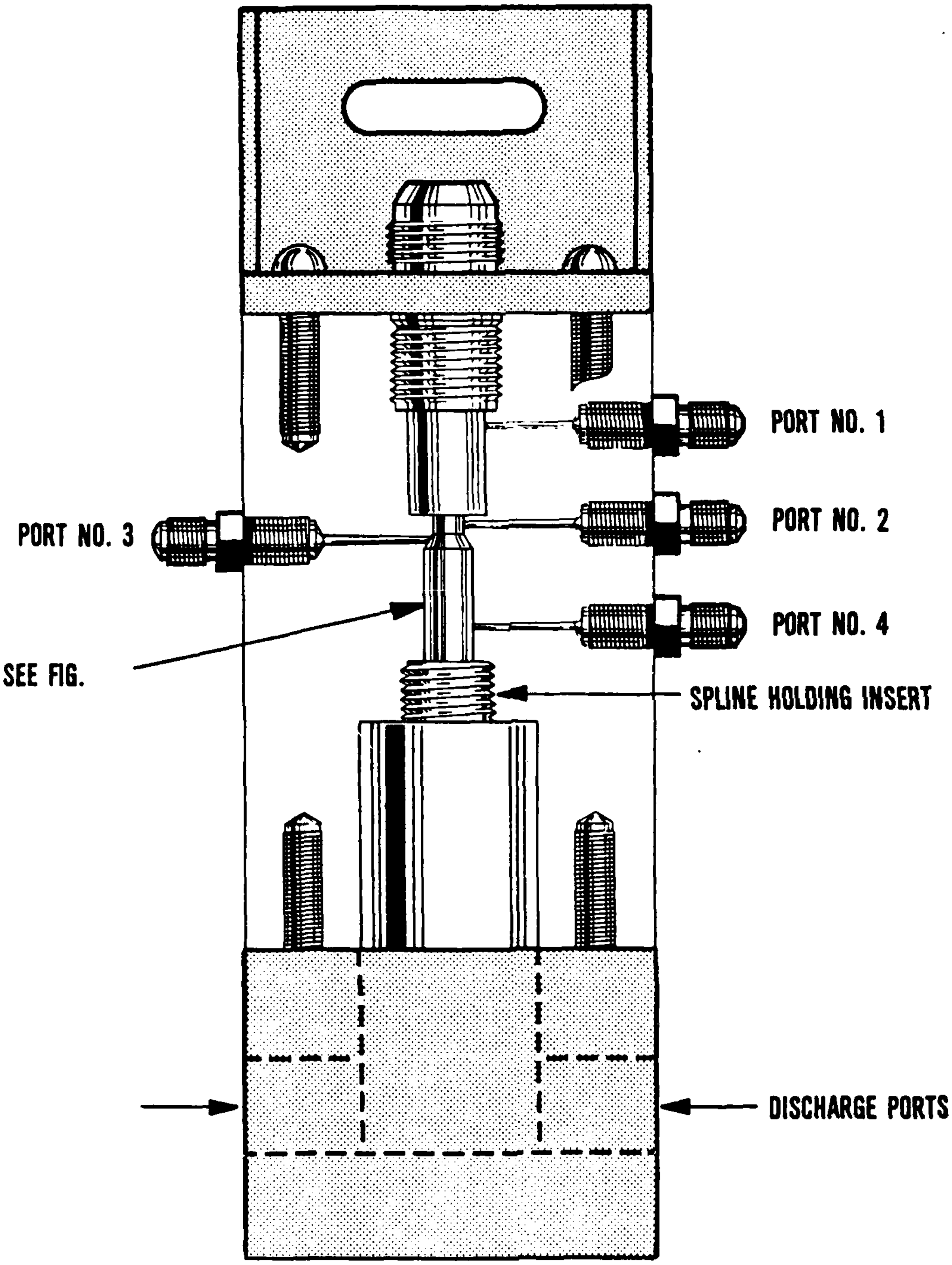



Figure 1-1. Overall View of the Plastic Orifice Tester.

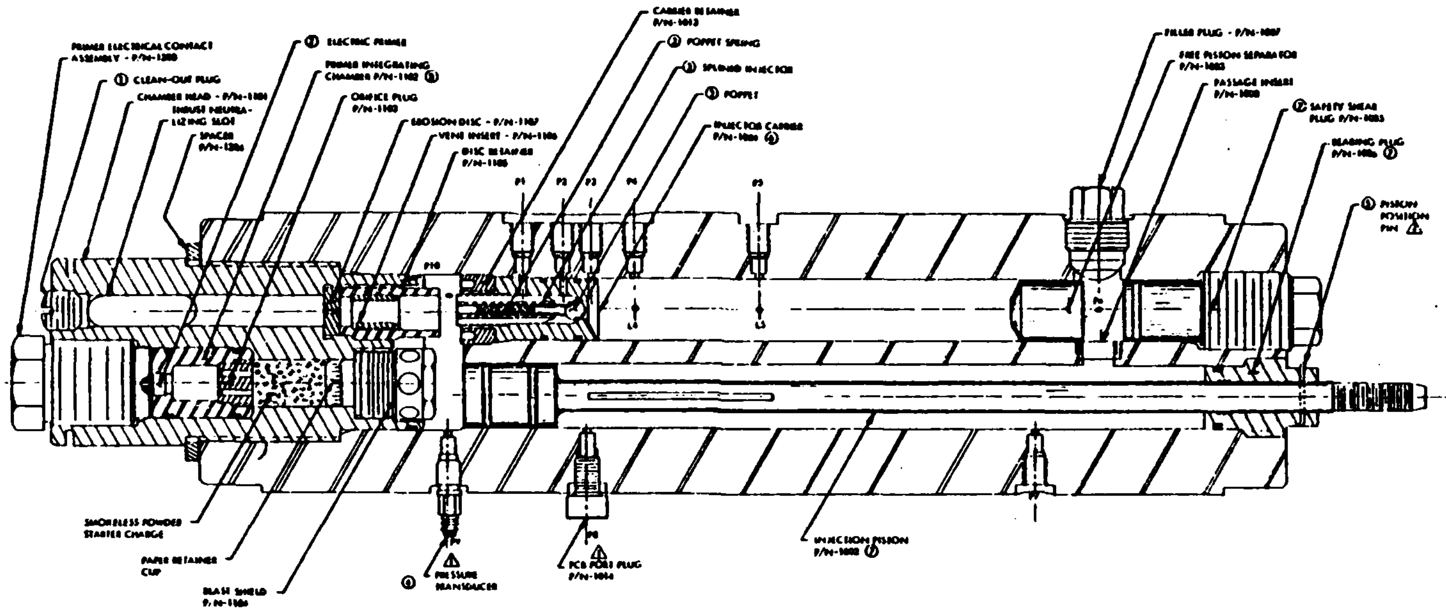



Figure 1-2. Assembly Drawing of Steel Orifice Tester.

The next design choice was how to pressurize the propellant reservoir while still keeping the relationship between the combustion chamber and propellant reservoir pressures comparable to that found in a regenerative test fixture. The solution adopted was to allow the combustion chamber to pressurize the propellant reservoir through a hydraulic multiplier piston. Accordingly, the steel test fixture is configured as a "U" tube, with the injector orifice and propellant reservoir in the upper bore of the "U" and the hydraulic multiplier piston in the lower bore. The upper part of the "U" is exposed to combustion chamber pressure. The working fluid in the "U" is water. The water is separated from the propellant reservoir by a free floating piston.

The inherent limitation of this approach is that its ability to properly and safely respond to rapid changes in combustion chamber pressure is marginal. Changes in combustion chamber pressure are immediately seen at the injector orifice as it is directly exposed to the combustion chamber. However, this same pressure change cannot create a comparable pressure rise in the propellant reservoir through motion of the injection piston until the pressure signal completes its passage around the "U" tube. This travel time is in excess of 0.5 msec. Thus, with this fixture, there is an ever present possibility of reverse flow of hot gases through the injector orifice in a way which would not occur in an actual regenerative gun fixture. However, it was intended that this test fixture simulate the operation of a large caliber artillery gun, for which the pressure rise times are long compared to 0.5 msec. It was therefore planned to carefully control the combustion chamber pressure rise rate.

The concept and specification for this test fixture were determined by General Electric Ordnance Systems Division. The detailed design, fabrication, and initial checkout testing were performed by the Princeton Combustion Research Laboratories under subcontract to GEOSD.

## Section 2

## TESTS WITH THE PLASTIC FIXTURE

### 2.1 INJECTOR DESCRIPTION

The first injector orifice chosen for investigation was the ball check injector. A diagram of the orifice investigated is shown in Figure 2-1. It consists of a threaded spline, a spring, and a ball bearing with a shaft braised to it. A picture of these parts, disassembled, is shown in Figure 2-2. A drawing of this injector is shown in Figure 2-3. The splines are tapered on their upstream side. The spring resides in a bore in the spline and acts against the shaft to keep the ball pressed against its seat. In normal operation it opens when the pressure acting on the upstream face of the ball is sufficient to overcome spring pressure. This orifice was at one point used extensively in General Electric's Ordnance Systems Division (OSD) Independent Research and Development (IR&D) program and was characterized by a low inferred orifice flow coefficient and a high incidence of piston reversals. Analysis of the data pointed to the injector as the most likely culprit. It therefore appeared to be a good candidate for studying how not to design an injector orifice.

### 2.2 TEST DESCRIPTION AND DATA ANALYSIS

Considerable data was generated in water flow testing of the ball check valve in the plastic test fixture. The pressure drops were limited to 100 psi, and the flow rates to only 15 gallons per minute (GPM), values two orders of magnitude less than those typical of gun fixture firings. However, even at such low flow rates, the Reynolds number for this flow passage is well into the turbulent regime, although the relatively short length to diameter (L/D) ratio of the flow passage may not allow sufficient time for a fully turbulent flow to develop in the orifice. It must also be noted that the complex geometry of this orifice will promote the occurrence of flow separation and/or cavitation.

The tests were performed in two modes. In the first mode the ball was fixed at various axial positions and the flow rate varied through the available range. In the second mode the ball was allowed to float against the spring contained in this spline. Two sizes of flow meters were used, a 1/2 inch and a 3/4 inch. Photographs of five typical flow conditions are shown in Figures 2-4 through 2-8. In general, for the tests where the ball floated against the spring (see Figure 2-6), some oscillation of the ball was observed. Since this significantly complicates numerical data analysis, only the results from the fixed ball flow tests will be presented.

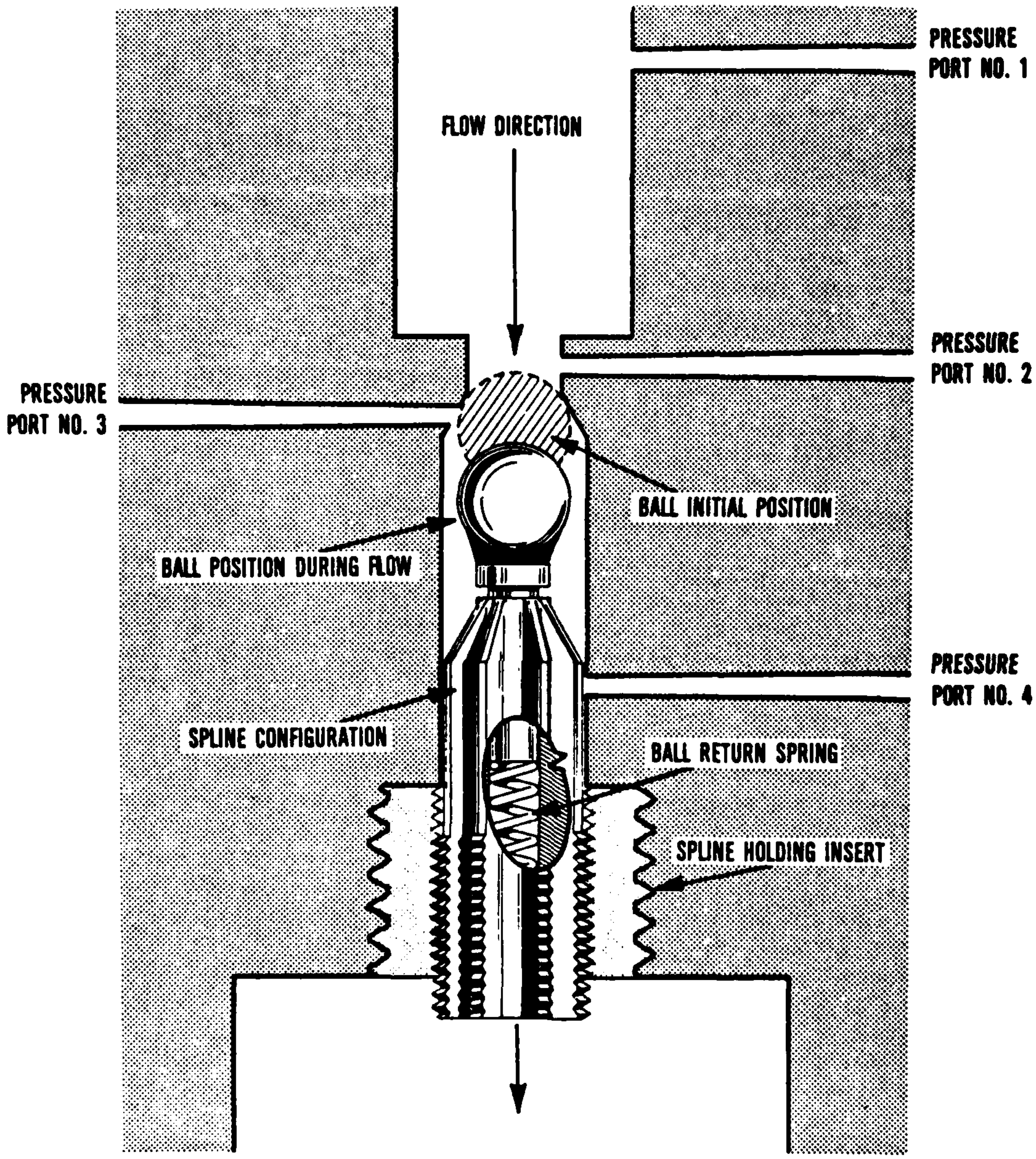



**Figure 2-1. Details of the Ball Check Injector.**

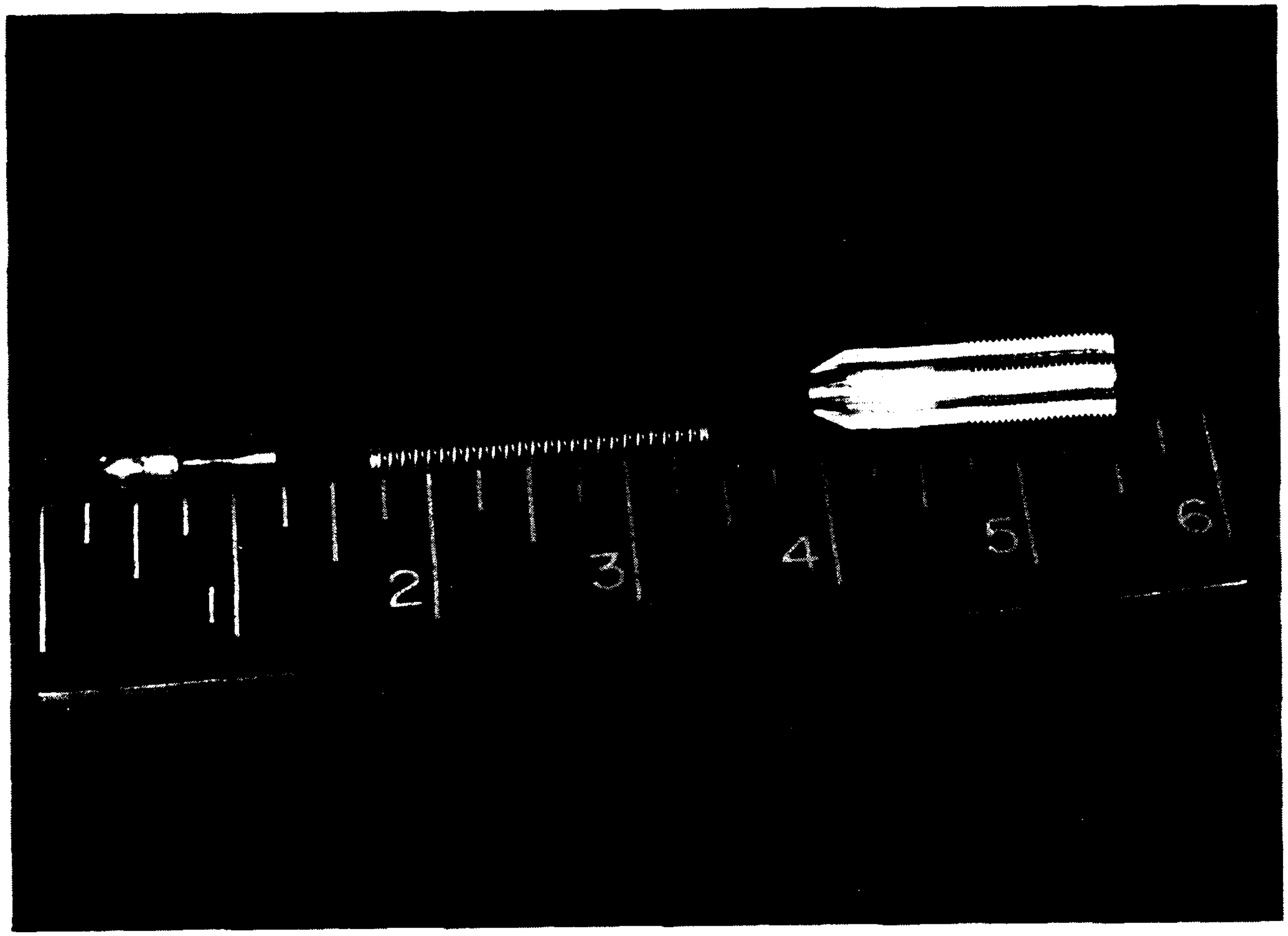



Figure 2-2. Picture of Ball Check Injector.

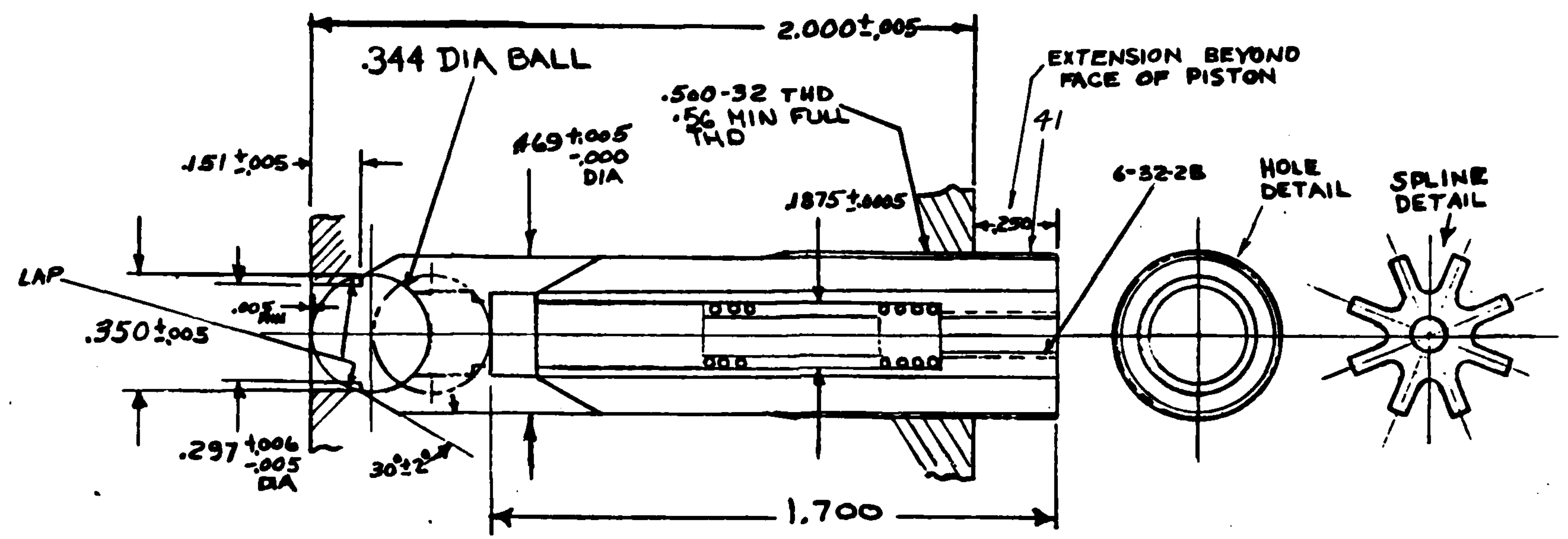



Figure 2-3. Drawing of Ball Check Injector.

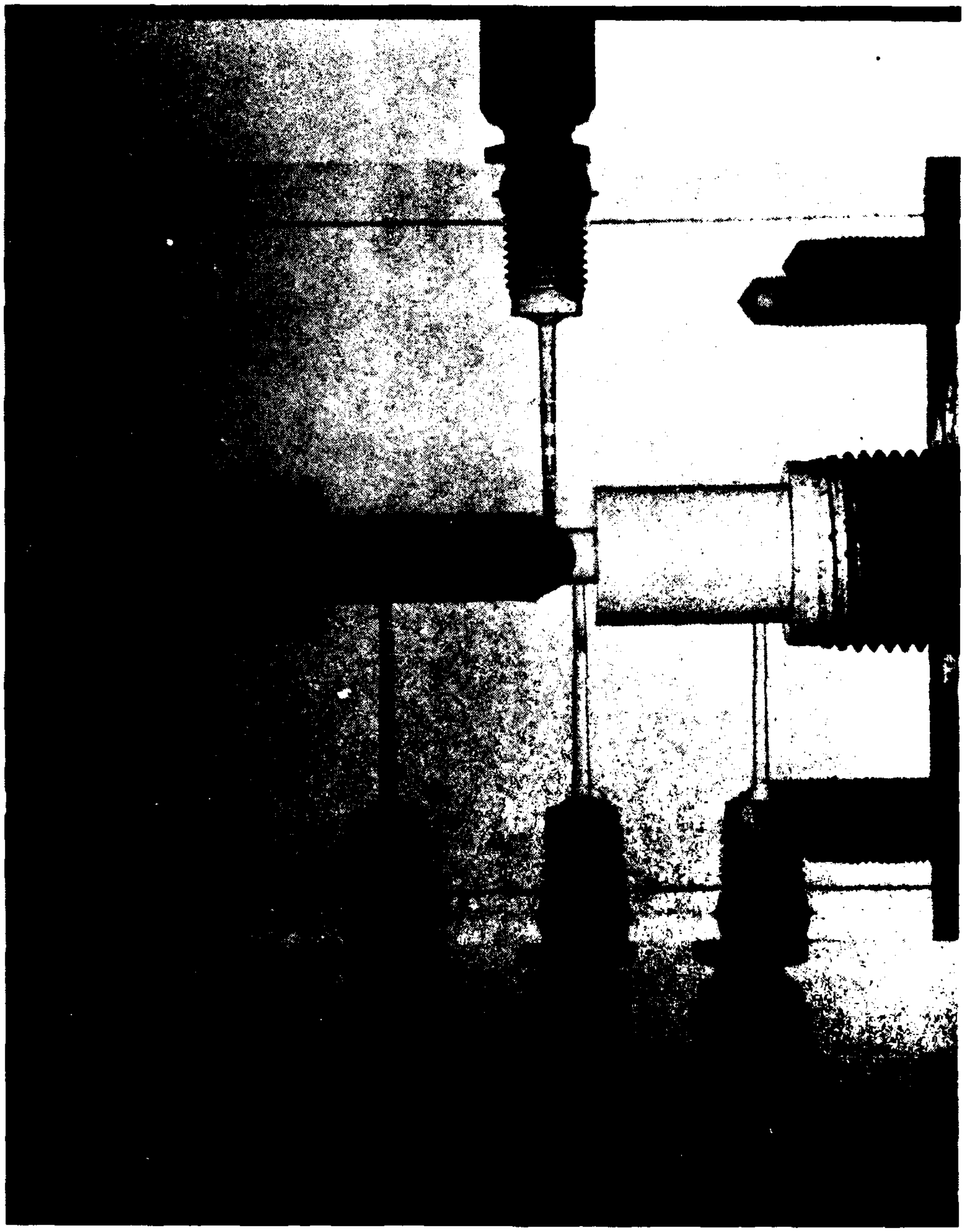


Figure 2-4. Flow Through Ball Check Injector, Plastic Orifice Tester, Ball Set 0.125 Inches from Shoulder, Flow Rate of 2.55 GPM, Inlet Pressure 29 psi. Diameter of Bore Holding Injector is Approximately 0.47 Inches.

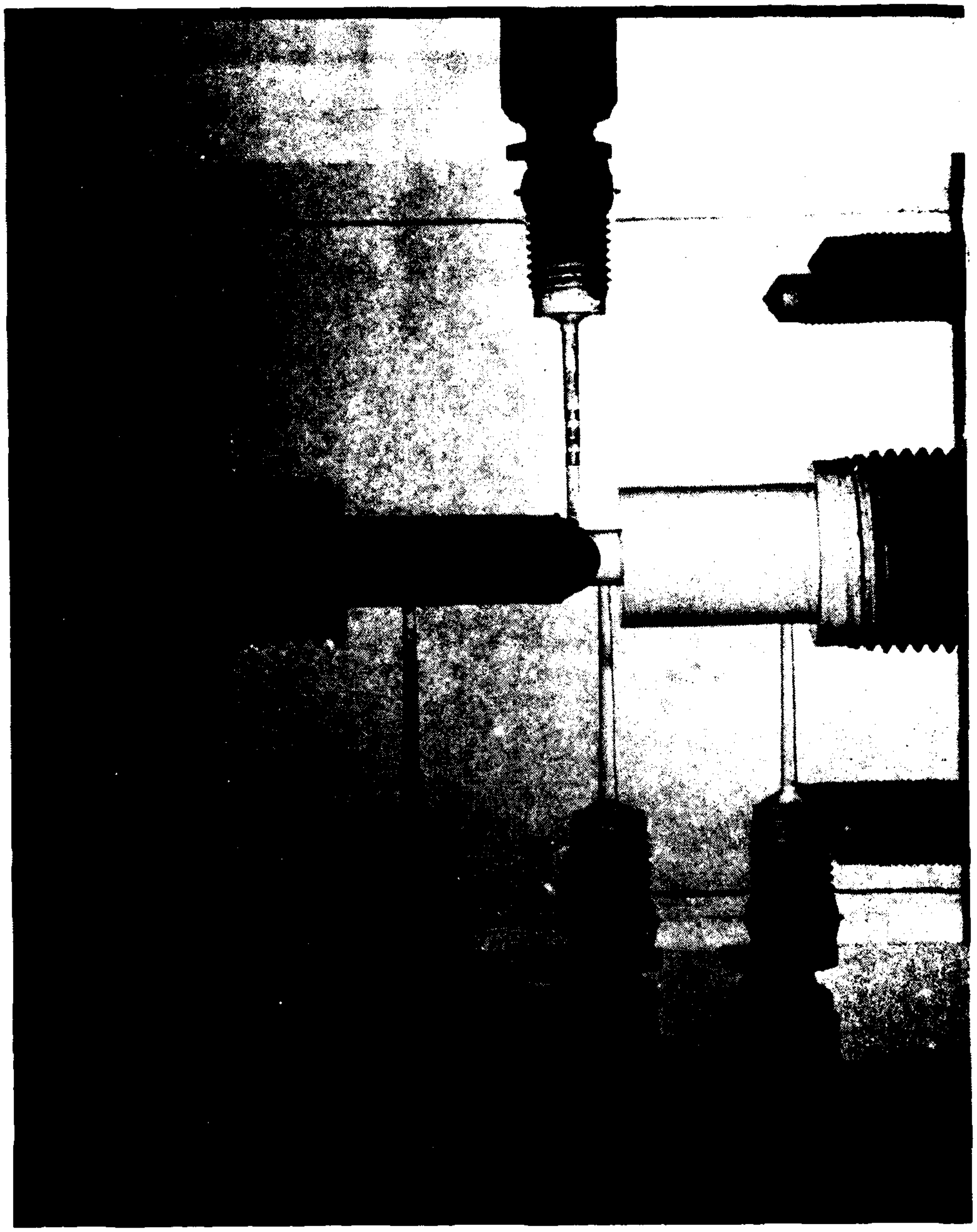


Figure 2-5. Flow Through Ball Check Injector, Plastic Orifice Tester, Ball Set at 0.125 Inches from Shoulder, Flow Rate of 3.2 GPM, Inlet Pressure of 56psi.

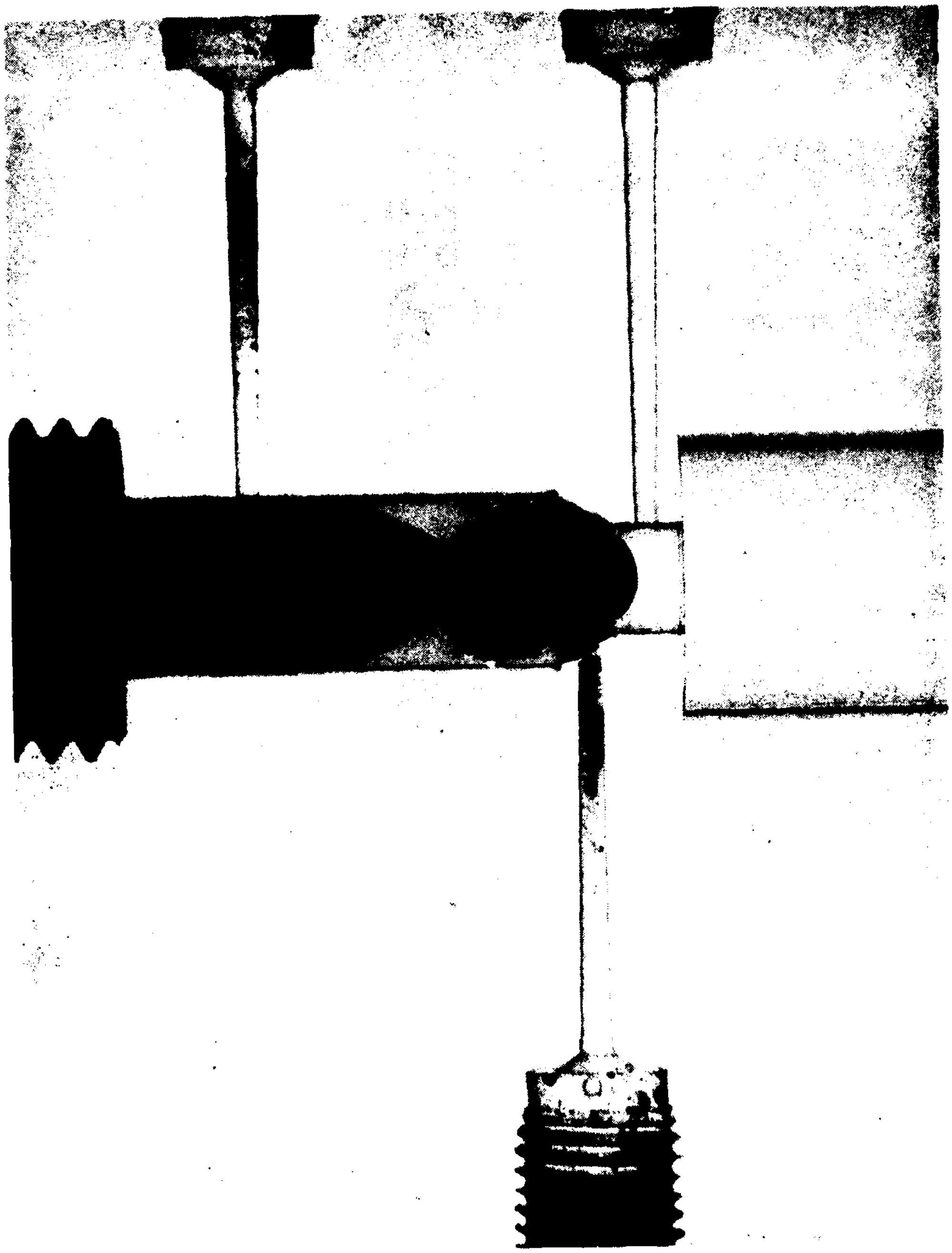


Figure 2-6. Flow Through Ball Check Injector, Plastic Orifice Tester, Ball Free Floating Against Spring, Flow Rate of 4.6 GPM, Inlet Pressure 60psi.

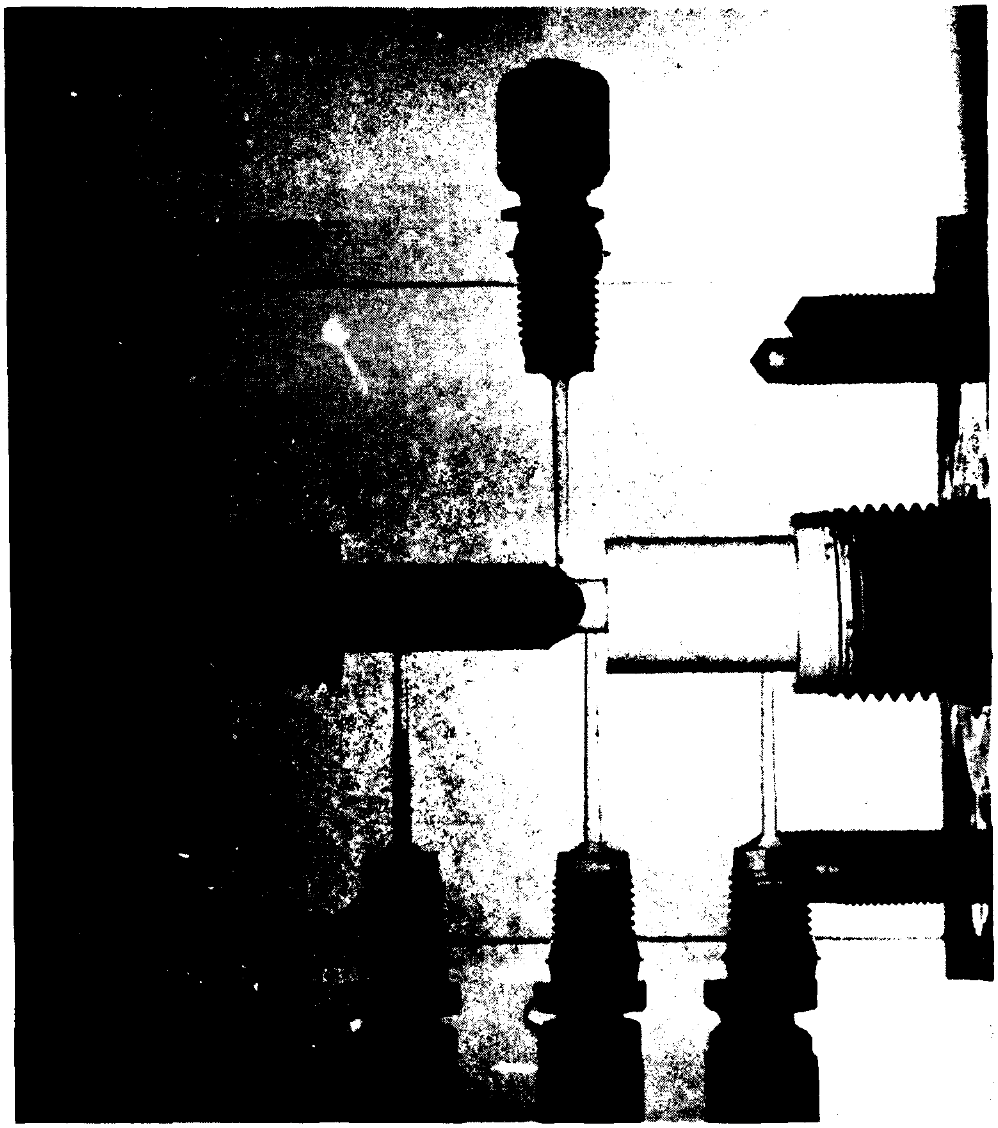


Figure 2-7. Flow Through Ball Check Injector,Plastic Orifice Tester, Ball Free Floating Against Spring, Flow Rate of 5.95 GPM, Inlet Pressure of 59psi.

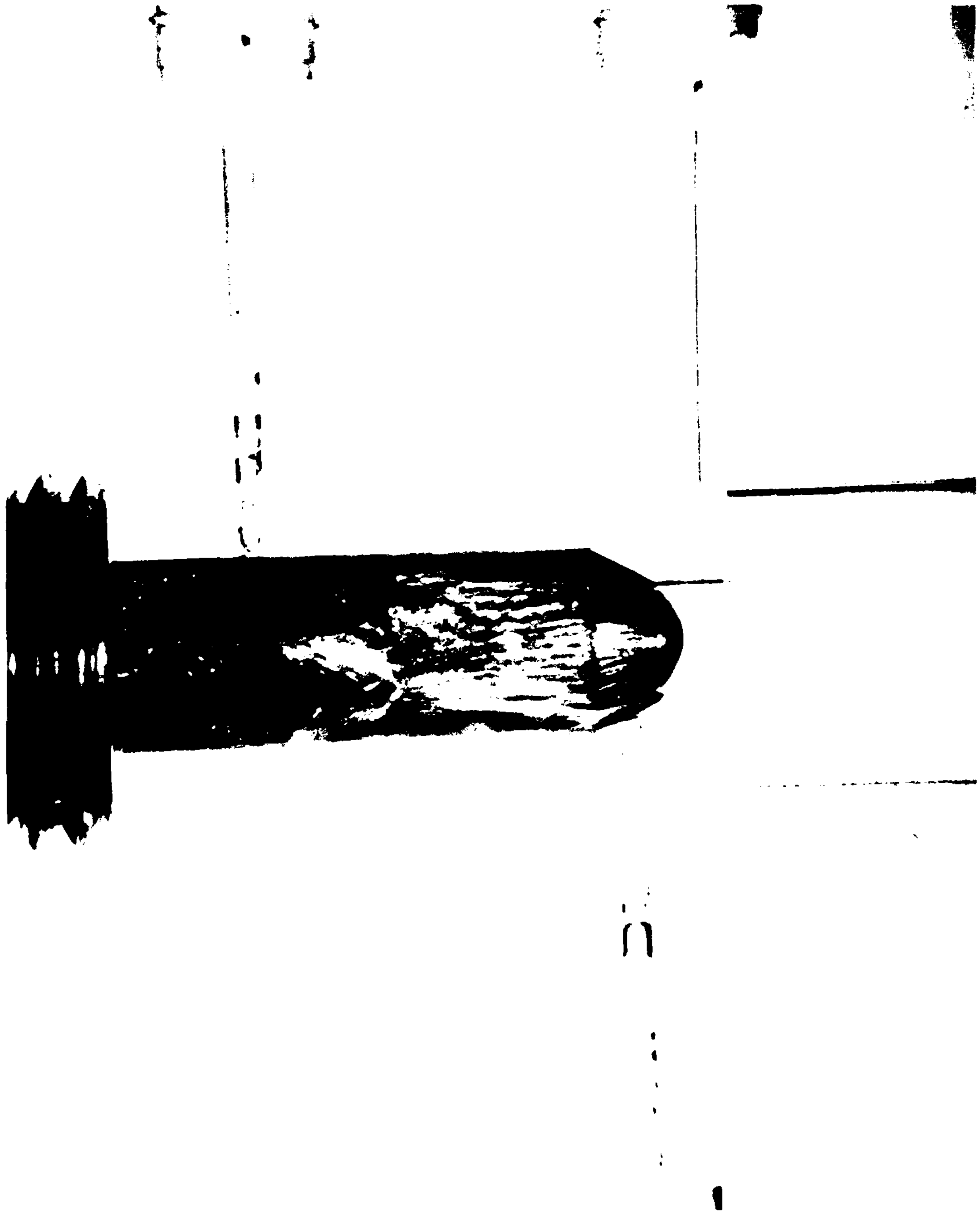


Figure 2-8. Flow Through Ball Check Injector

In order to better understand the flow in the orifice the data were reduced to a form more amenable to analysis. The total pressure of the flow at any point is a measure of the energy in the flow, and comparing the total pressure at various stations in the flow will provide clues as to the processes occurring between these stations. The total pressure was computed by calculating the flow velocity at each station assuming full flow through the available geometrical area. This, unfortunately, is a questionable assumption because, as has been mentioned earlier, this flow passage is likely to be subject to flow separation and cavitation. It should therefore not be surprising if some contradictions arise in the analysis of the data.

In order to better compare the flows at differing flow rates, the total pressure was normalized with respect to the dynamic pressure at the exit of the orifice. Several runs of identical, or nearly identical, initial conditions were analyzed on this basis and showed differing results. The following paragraphs discuss these results.

### 2.2.1 RUNS 3 AND 17

The reduced pressure data from these tests are shown in Figures 2-9 and 2-10. The tests were performed with the ball set at its maximum opening (0.1776 inches) which gave the highest discharge coefficient (0.7-0.95). Run 3 used the 1/2 inch flow meter with flows of 1-9 GPM. Run 17 used the 3/4 inch flow meter with flows of 4-15 GPM.

The circled numbers on the figures refer to the various pressure gauge taps (see Figure 2-1). Also shown on the figures is a value for the flow discharge coefficients, $C_D$, which was determined at the flow rate indicated in the figure.

Significant data overlap exist between these runs but the data do not repeat. Run 3 yields a consistently higher $C_D$ than Run 17. The form of the pressure profiles is strikingly different, most notably the pressures at stations 2 and 3. When these pressures are reversed, the match is closer. It may be possible that the data were mislabeled during recording. These traces were therefore interchanged in Figure 2-10. The measured static pressures, together with the calculated total pressures and velocities at one flow rate, are shown in Figure 2-11.

### 2.2.2 RUNS 11 AND 12

The data from these tests are shown in Figures 2-12 and 2-13. In these runs, the ball was fixed to give an opening axial clearance of 0.0995 inches. Again, the only intentional difference was that Run 11 had the 1/2 inch flow meter at flows of 1-9 GPM while Run 12 used the 3/4 inch flow meter at flows of 4-13 GPM. Run 11 yields a higher $C_D$ through the whole range of overlap. Except at station 4, the reduced pressures differ significantly. An unaccountable cusp exists in the reduced pressure at station 1 for Run 12.

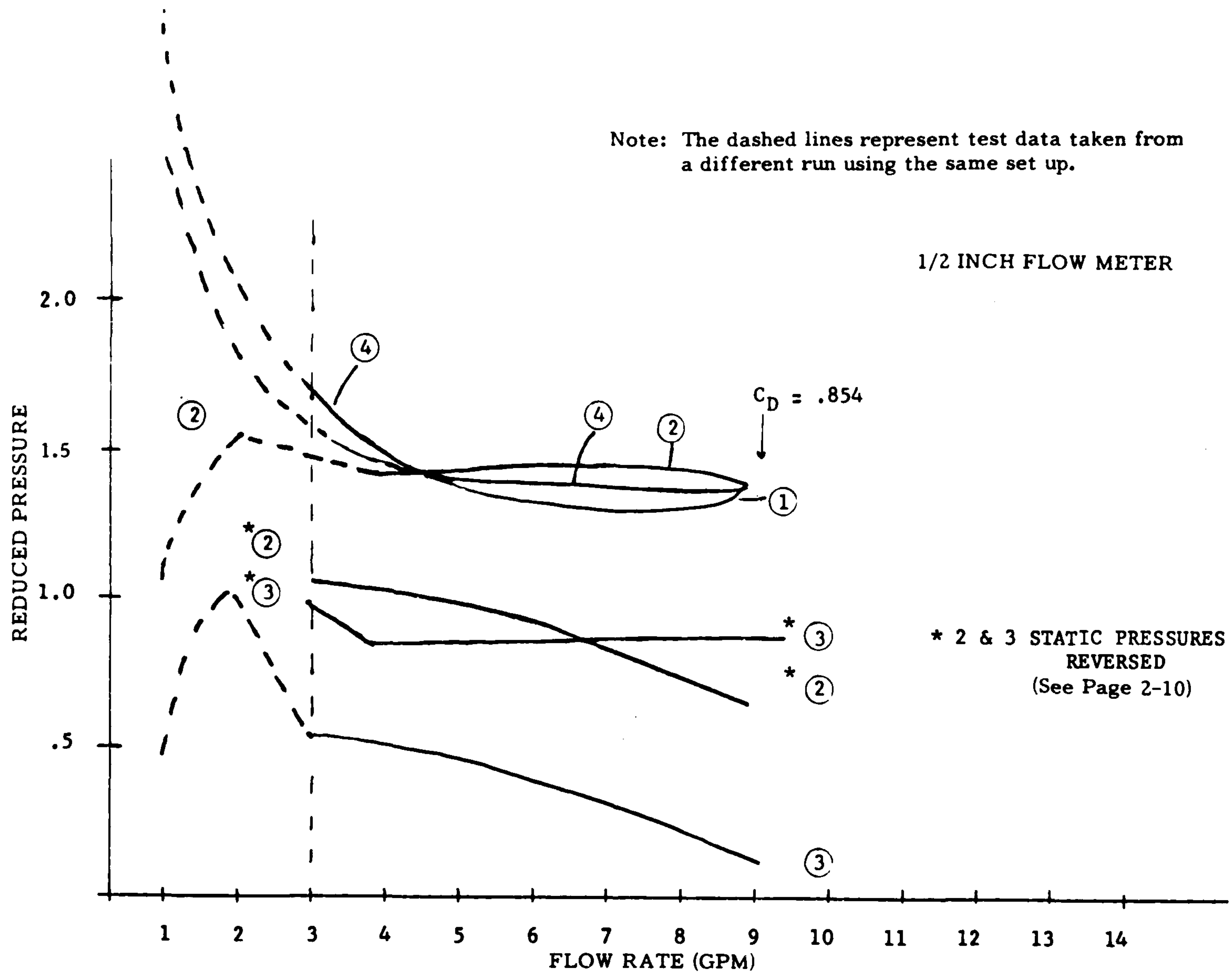
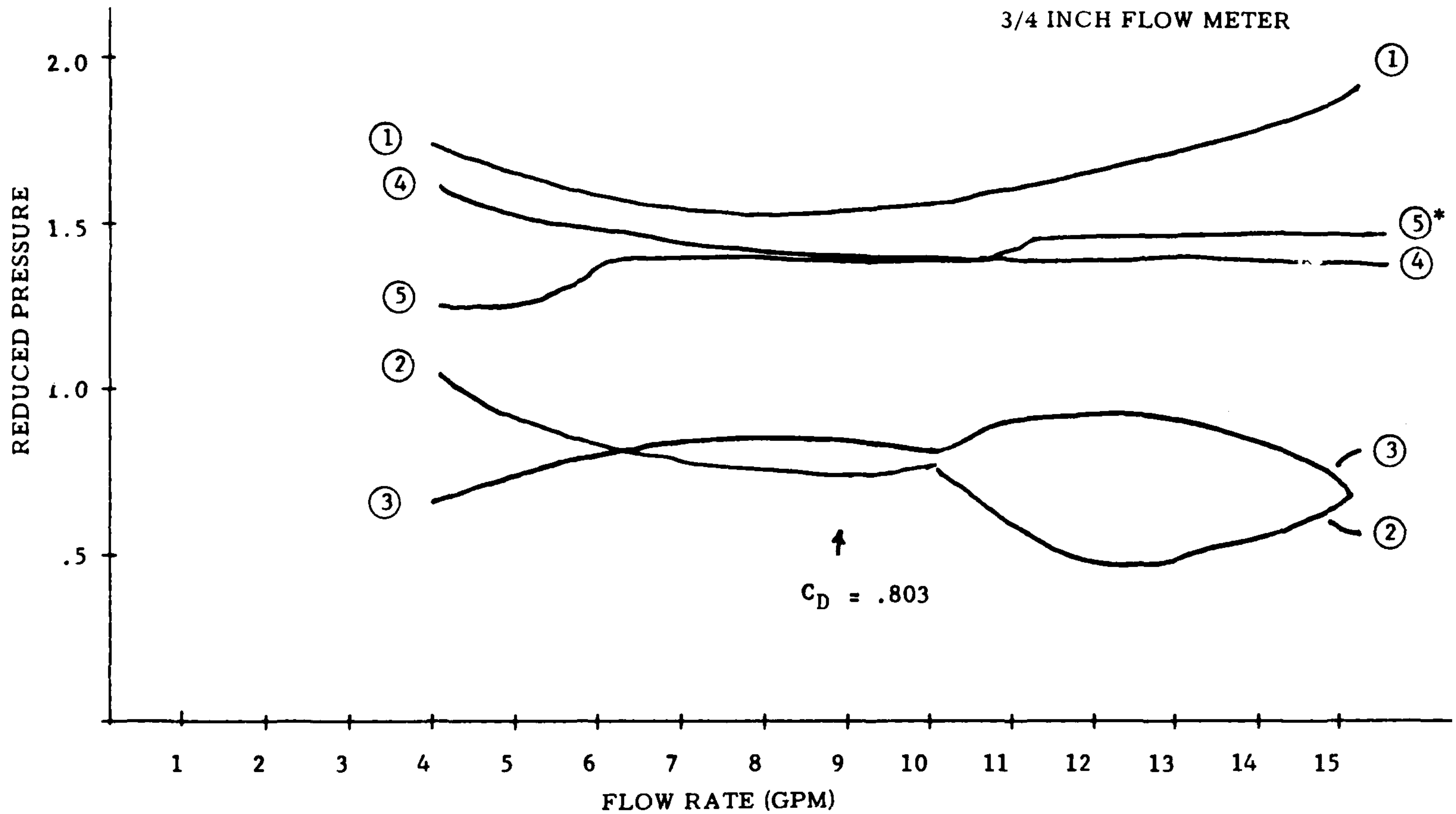
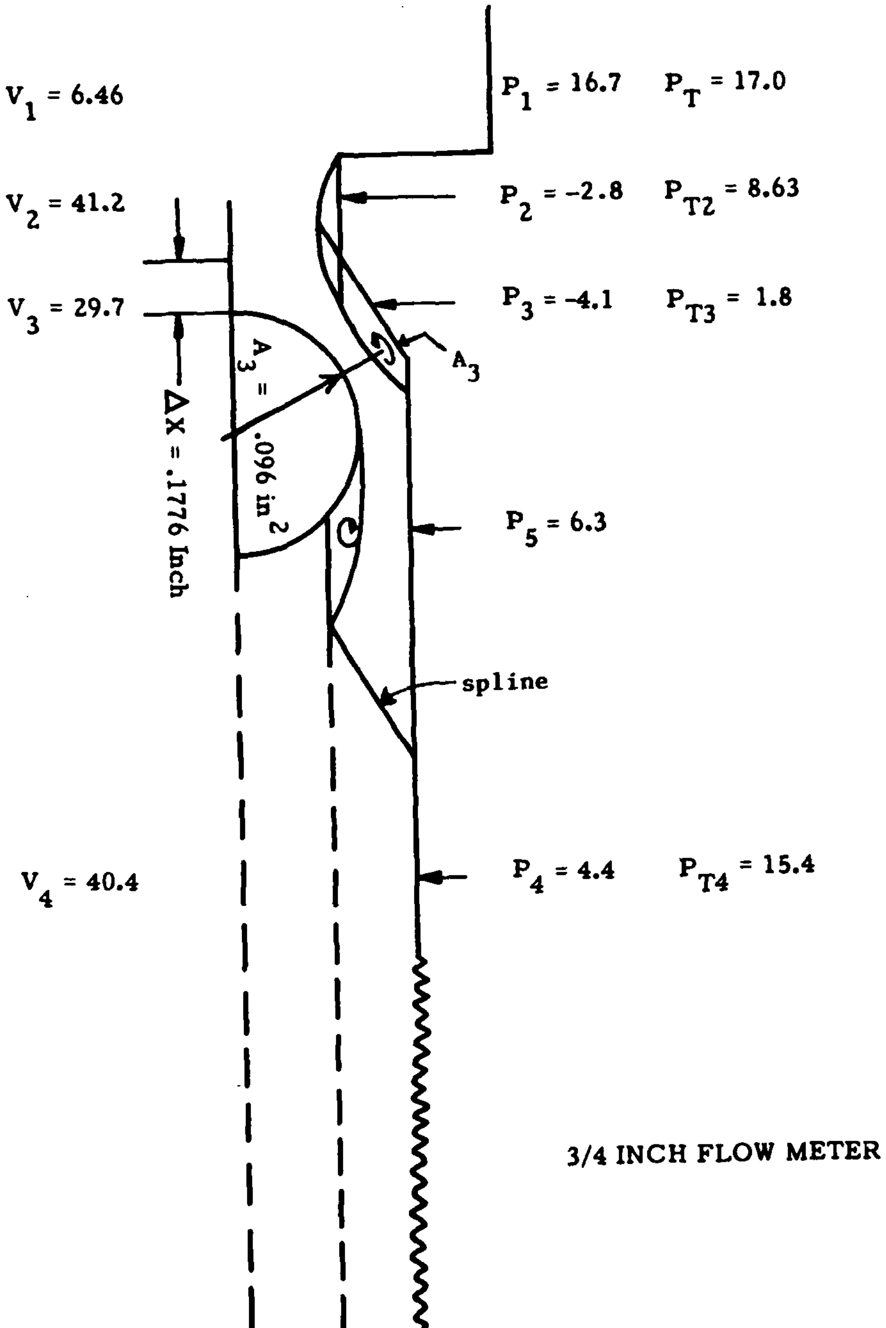



Figure 2-9. Reduced Pressure Data from Run 3 (Ball Open 0.1776 Inch).




*NOTE: Station 5 is an additional gauge location added later in the region of just downstream of the ball (see Figure 2-11).

Figure 2-10. Reduced Pressure Data from Run 17 (Ball Open 0.1776 Inch).




Note: Pressure is in psig and Velocity is in ft/sec.
$C_D$ =.9347

Figure 2-11. Analysis of Flow in Injector for Run 17 at a Flow Rate of 8.9 GPM.

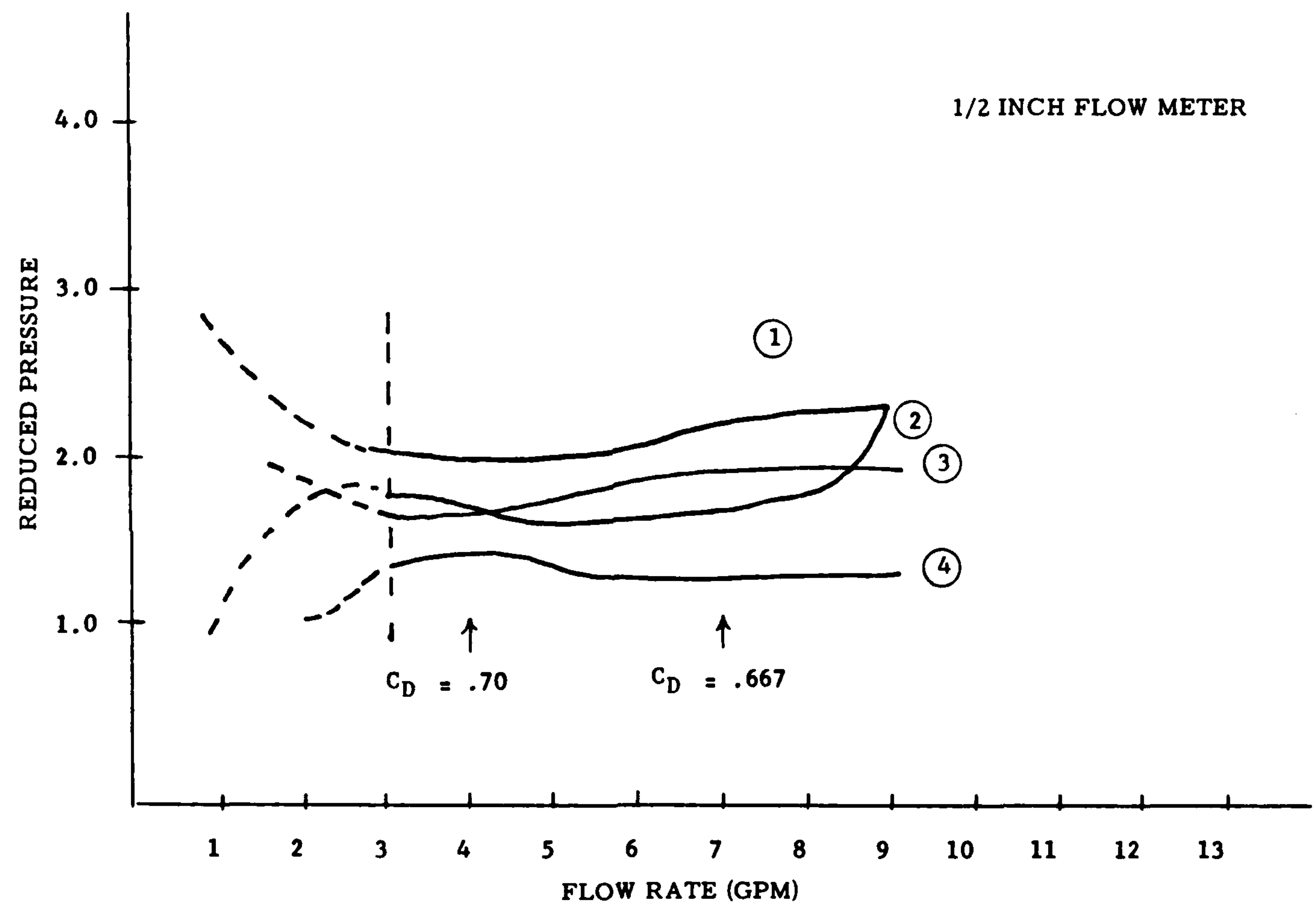



Figure 2-12. Reduced Pressure Data from Run 11 (Ball Open 0.0995 Inch).

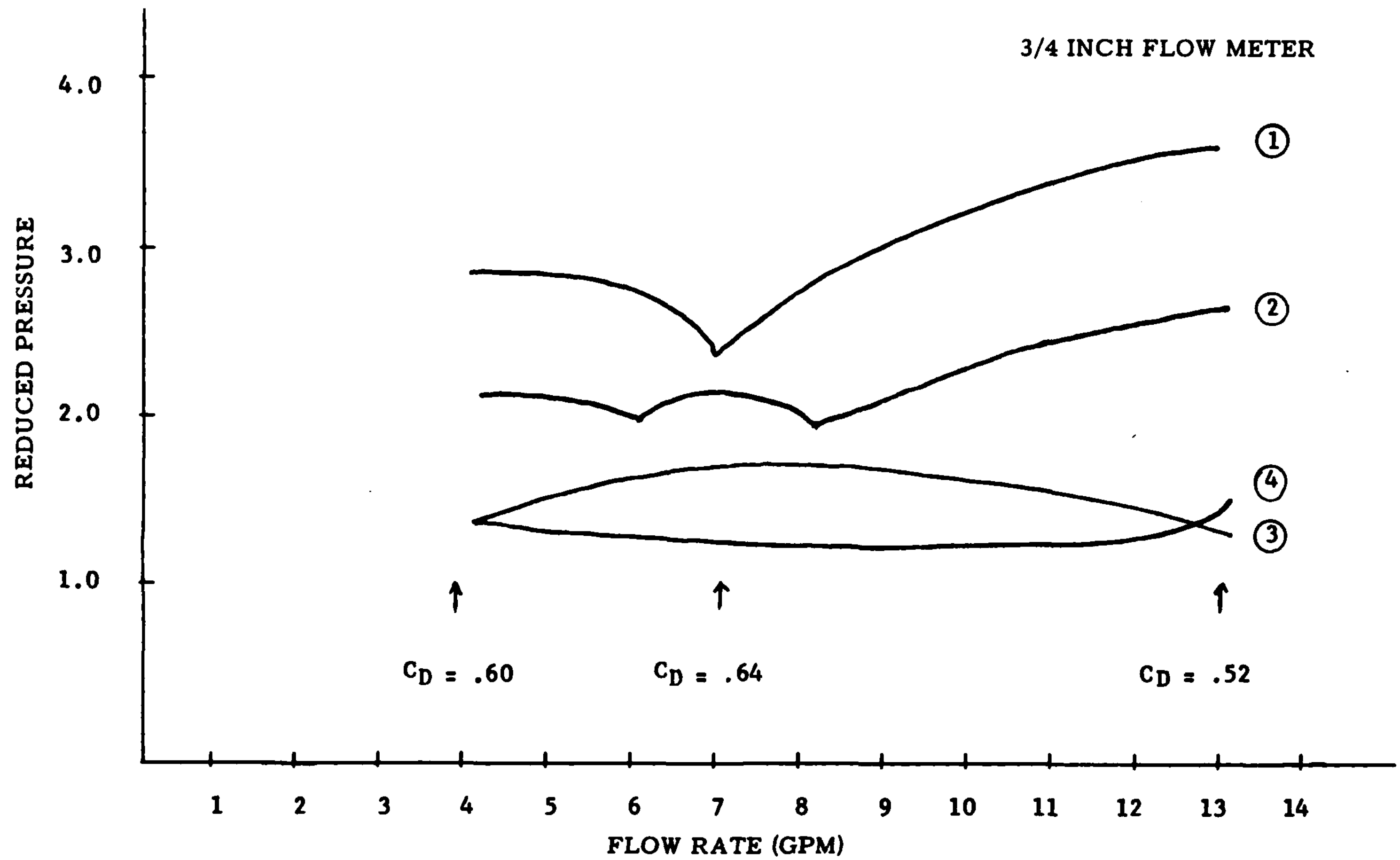



Figure 2-13. Reduced Pressure Data from Run 12 (Ball Open 0.0995 Inch).

### 2.2.3 RUNS 9 AND 14

The data from these tests is shown in Figures 2-14 and 2-15. Run 9 had a ball opening of 0.0526 inch and used the 3/4 inch flow meter. Run 14 had a larger opening of 0.0604 inch and used the 1/2 inch flow meter. The comparison of these two runs yields the greatest paradox of the series. In all the previous runs, use of the 1/2 inch flow meter gave the highest $C_D$. This is not so in this case, where use of the 3/4 inch flow meter, coupled with a lower flow area, gave the higher $C_D$.

Significant structure is evident in the pressures recorded in Run 14. Stations 2 and 3 show remarkable similarity in this structure. Station 4 shows an apparent mirror image of station 3 and station 1 shows a highly smoothed version of station 2. To investigate this structure further, Run 15 (identical to Run 14) was also analyzed. The data from this test are shown in Figure 2-16. The trends were the same but a different structure appeared. When the average values from Runs 14 and 15 are plotted (see Figure 2-17), the structure becomes much smoother. Stations 1 and 2 track each other and stations 3 and 4 are mirror images of each other.

The pressure at station 4 shows a large dip in pressure, which is difficult to understand. This cannot be explained by one dimensional single phase flow. It may be evidence for local cavitation entering the spline (reference page 2-23 for discussion). The two-phase flow in the splines would have a lower density and would require a higher velocity as the flow proceeded through the splines. As the vapor returned to the liquid phase, the velocity would decrease and the pressure would rise to the back pressure.

The measured static pressures, and calculated total pressures and velocities, for one flow rate are shown in Figure 2-18.

Run 9 also exhibits some paradoxes. The first is that the calculated total pressure at station 3 (see Figure 2-14) exceeds the supply pressure up to a flow of 9 GPM. This has to be an experimental error since no system can show an increase in total head without the addition of work. Either the pressure measured is too high or the area used is too low. The second is that the pressure at station 3 shows an increasing suction up to 10 GPM. After this point it remains at about 12 psi below atmospheric pressure (approximately 3 psia). The pressure at station 3 could not have been much lower. This only leaves the flow area for the source of major error. Therefore, the opening of the ball must have been larger (at least 15%) than the stated 0.0526 inches. Returning to the pressure at station 3, the very low and nearly constant static pressure above 10 GPM suggests cavitation at this location. The static pressure at 4 shows a sudden jump at this flow rate and also holds constant as the "choking" effect of the cavitation propagates down stream.

The measured static pressures, and calculated total pressures and velocities for one flow rate are shown in Figure 2-19.

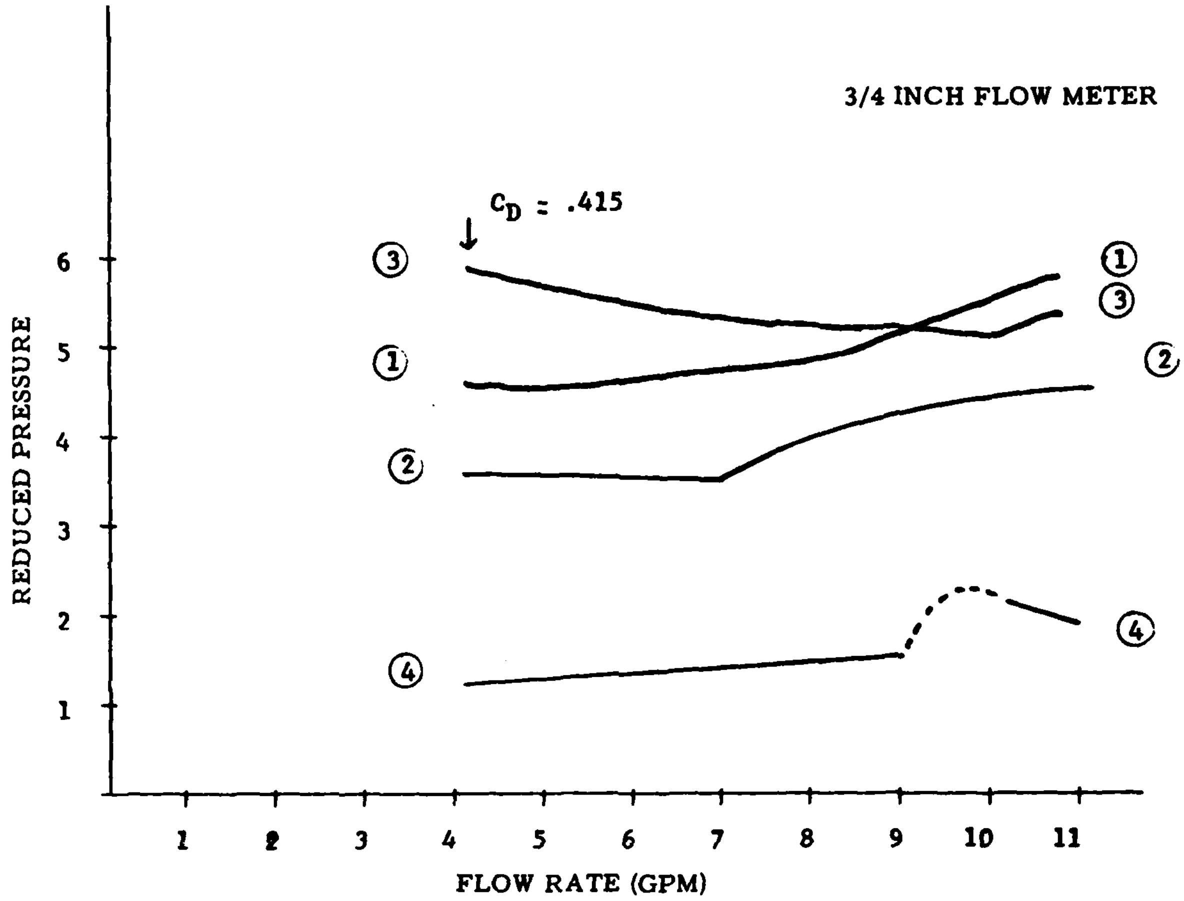



Figure 2-14. Reduced Pressure Data from Run 9 (Ball Open 0.0526 Inch).

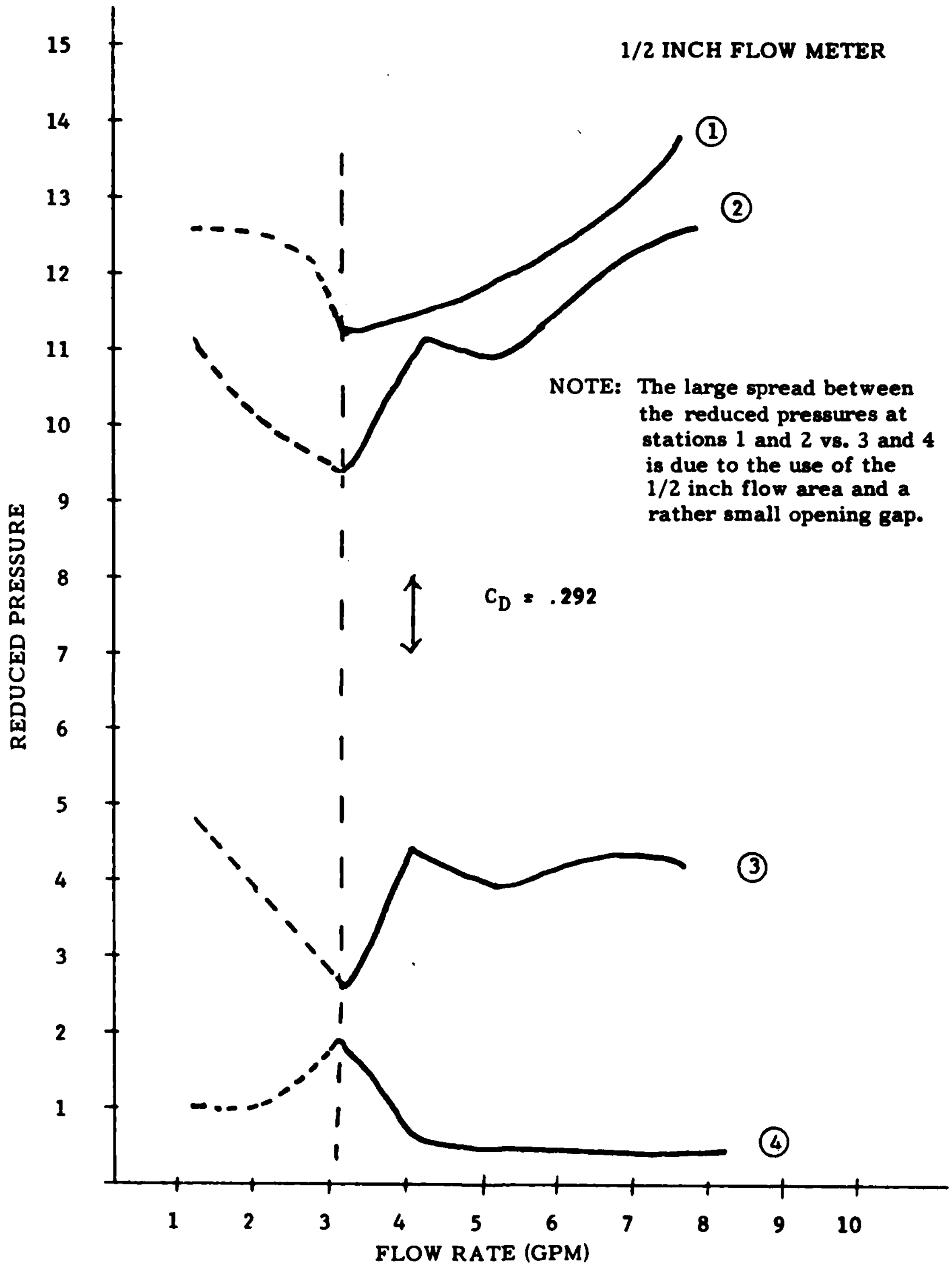



Figure 2-15. Reduced Pressure Data from Run 14 (Ball Open 0.0604 Inch).

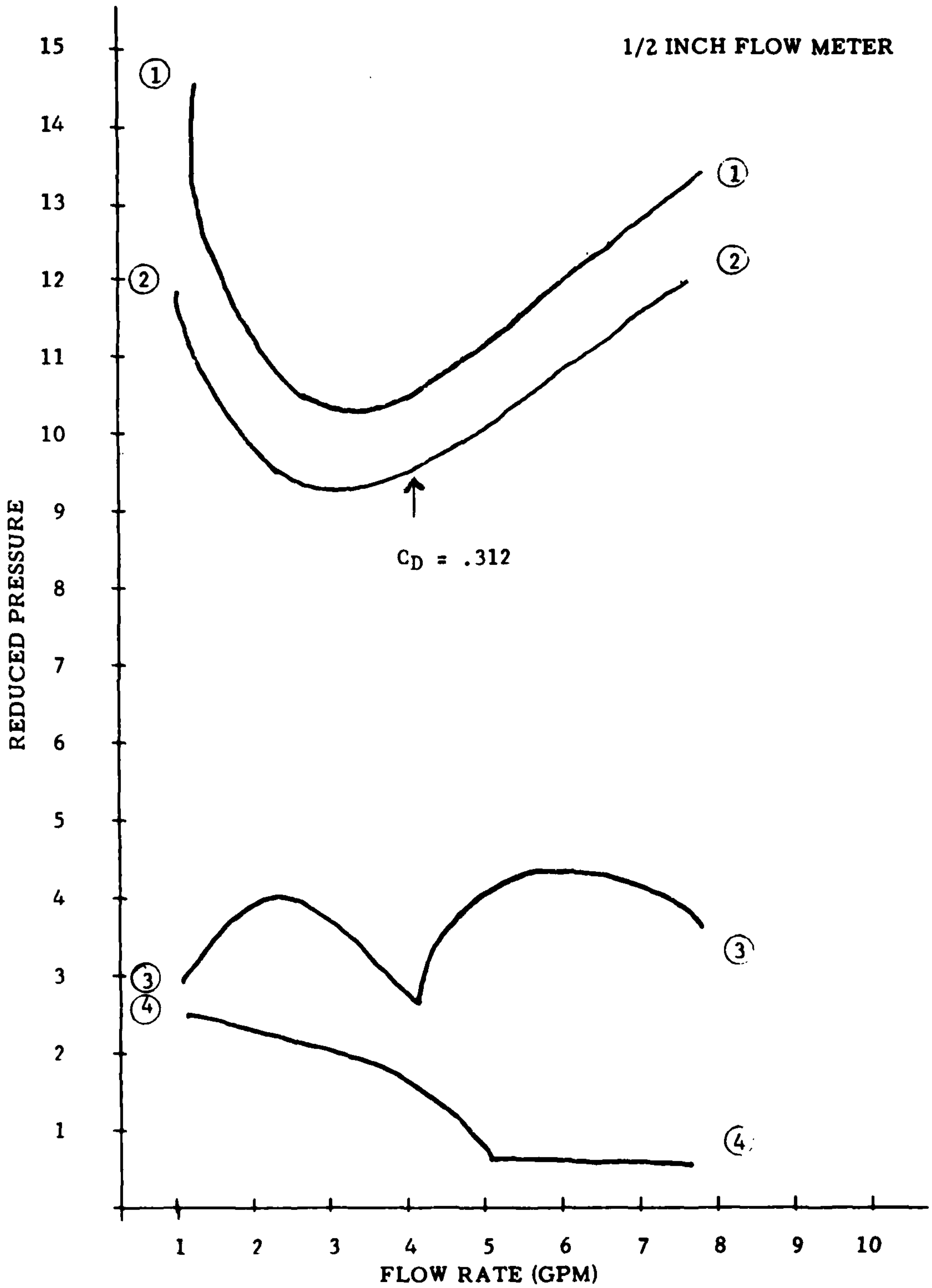



Figure 2-16. Reduced Pressure Data from Run 15 (Ball Open 0.0604 Inch).

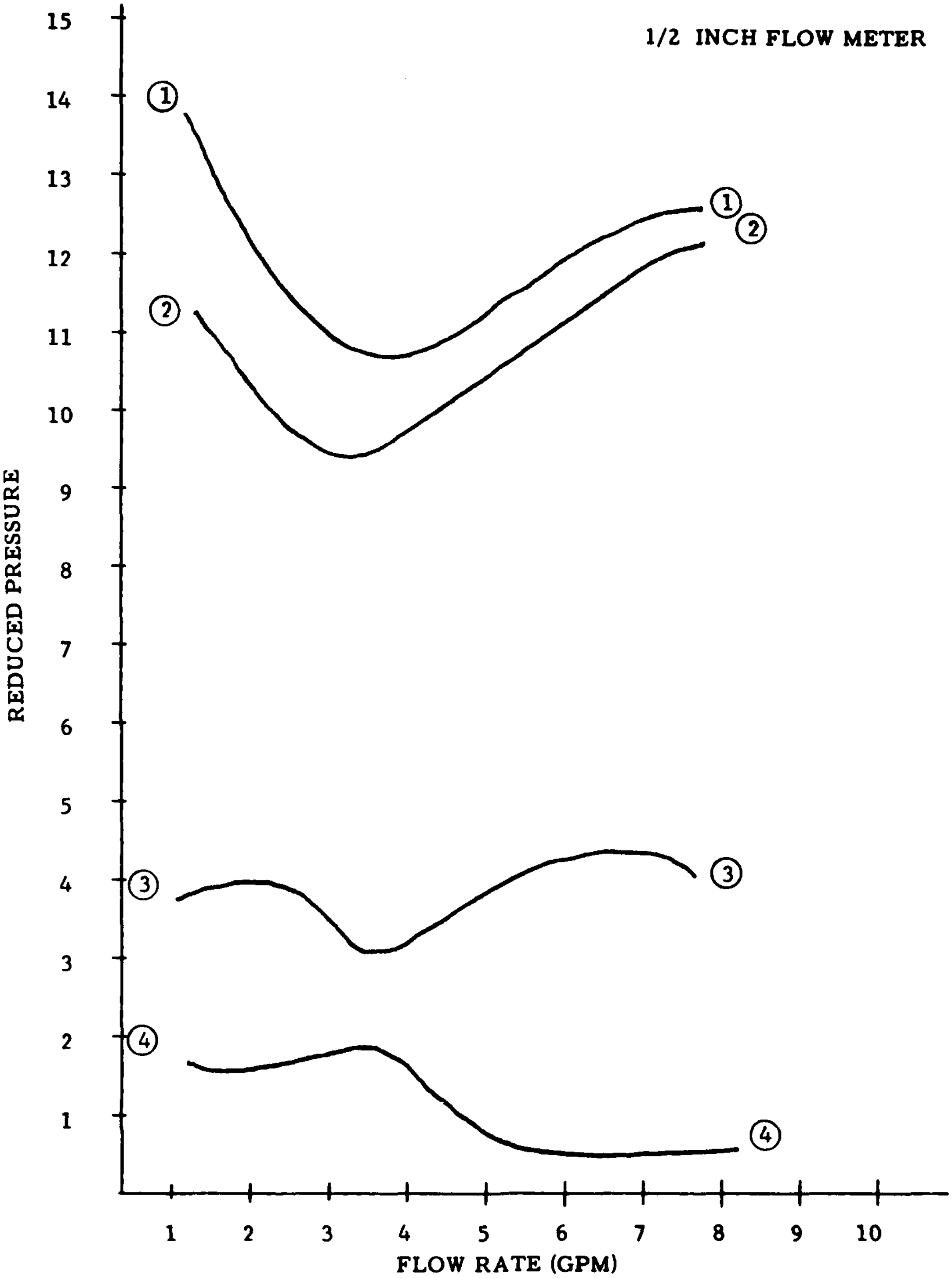
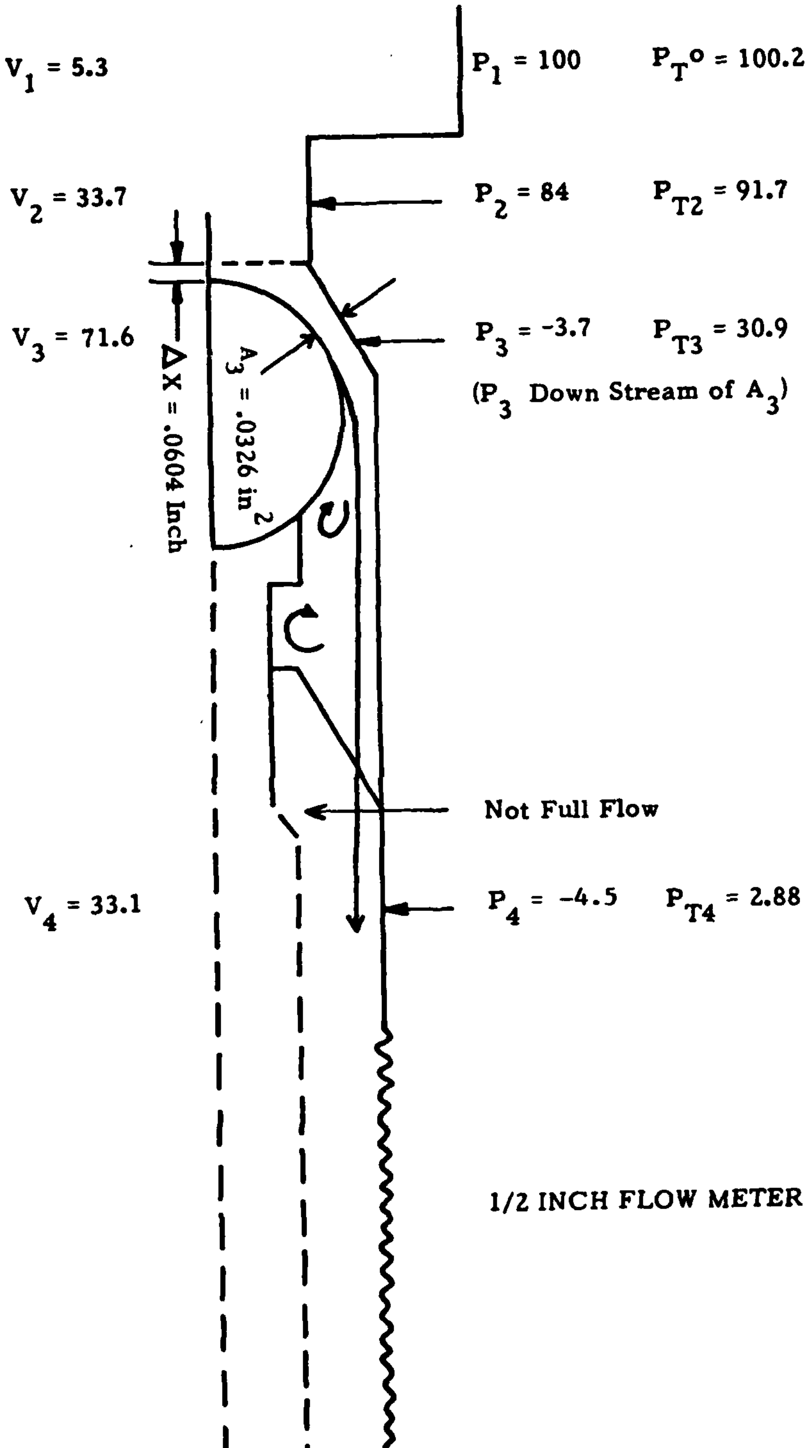


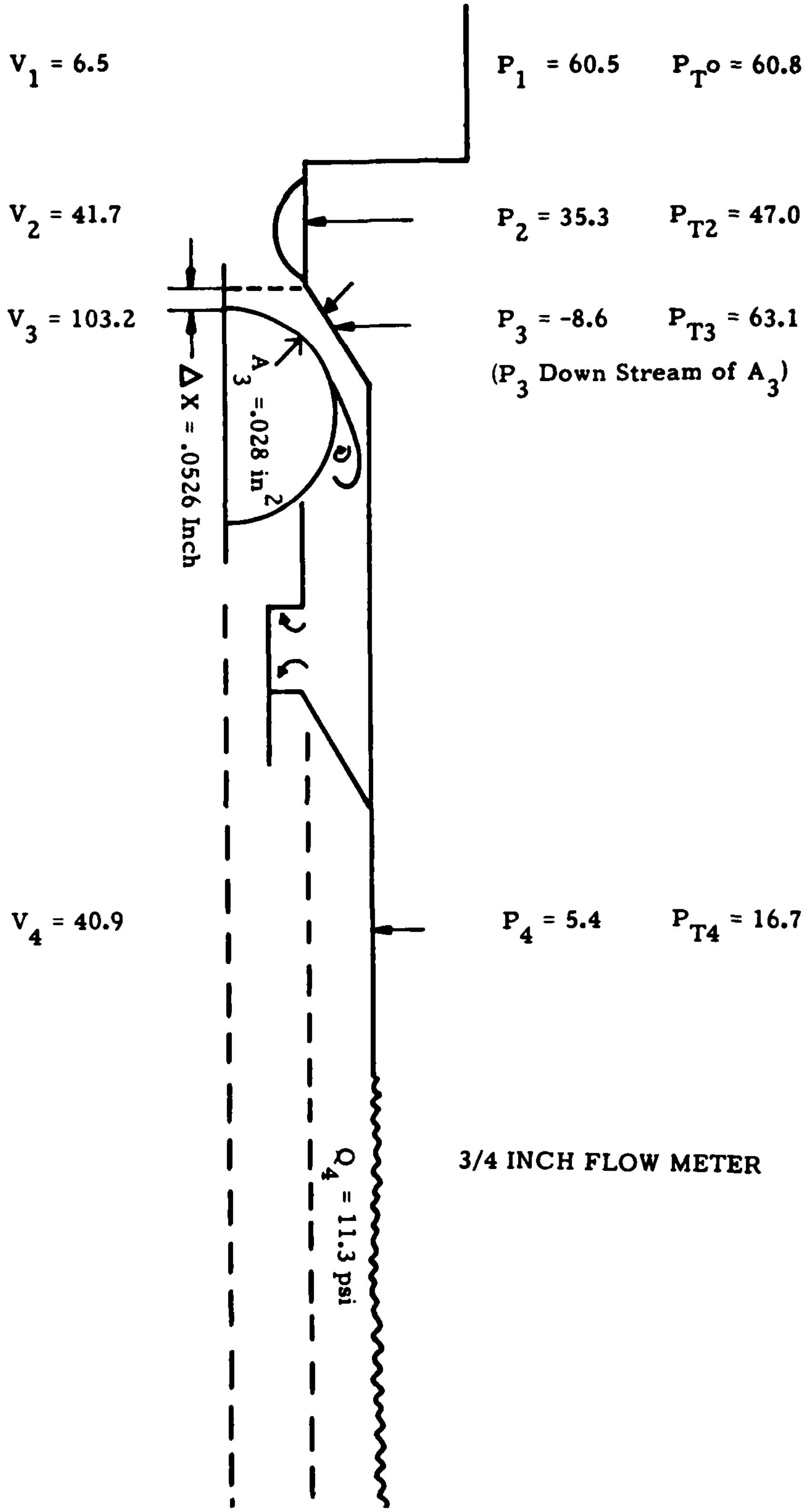

Figure 2-17. Reduced Pressure Data from Average of Runs 14 and 15 (Ball Open 0.0604 Inch).




Note: Pressure is in psig and Velocity is in ft/sec.

$C_D = .2623$

Figure 2-18. Analysis of Flow in Injector for Run 14 at a Flow Rate of 7.28 GPM.




Note: Pressure is in psig and Velocity is in ft/sec.
$C_D$ = .4466

Figure 2-19. Analysis of Flow in Injector for Run 9 at a Flow Rate of 9.6 GPM.

## 2.3 DISCUSSION

The flow in the ball check injector is complex. Analysis of these data indicate that the flow is strongly three dimensional and at times multi-phase. The instrumentation also interferes with the data. The flow meter is located upstream of the initial inlet. Residual swirl left in the flow by the 3/4 inch flow meter is probably greater than for the 1/2 inch since it is larger in diameter. When the flow enters a small diameter section, more energy is tied up in the swirl component (due to the conservation of angular momentum) of the flow requiring a larger pressure differential for a given flow rate. The flow passages are highly curved and abrupt changes in area occur. As was mentioned earlier, this calls into question any calculations based on assumed one dimensional flow (i.e., the use of Bernoulli's equation). The pressure ports are large with respect to the flow passage dimensions and at station three, (Figures 2-1 and 2-18) are not even perpendicular to the surface. As a result, the measured pressures cannot be taken as accurate indications of the local static pressure.

The analysis indicates that, in several of these tests, cavitation is certain to be occurring at a number of points in the flow. A brief discussion of cavitation is presented here. By definition, cavitation occurs whenever the local pressure drops below the vapor pressure. When this occurs, sufficient liquid flashes to vapor to prevent the pressure from dropping further. The flow becomes a two phase flow until the pressure rises back above the vapor pressure and the vapor is driven back into the liquid phase. The rate at which vapor returns to liquid is fast, but not instantaneous. If the static pressure at every point in the flow were known, the existence of cavitation could be easily determined (occurring when the static pressure is less than or equal to the vapor pressure). In real systems, the pressure can only be measured at discrete points remote from the potential site of cavitation.

In hydraulics, a cavitation parameter $\sigma$ is defined by:

$$\sigma = \frac{P_S - P_V}{1/2 \ \rho V^2 + P_S}$$

where $P_V$ is the vapor pressure, $\rho$ is the density, $P_S$ is the static pressure (absolute) and V is the flow velocity. All these parameters must be measured as close to the potential cavitation site as possible. The denominator in the above expression is the total pressure.

If $P_S$ is an accurate measure of the pressure at the site, then, for flows which can be considered one dimensional, cavitation will occur when $\sigma = 0$. In more complex, multi-dimensional flows cavitation will also occur at $\sigma > 0$, and in extreme cases, with a highly complex and sharp flow geometry, cavitation will occur when $\sigma$ approaches 1. The value of $\sigma$ where cavitation first occurs, $\sigma_C$, is usually determined experimentally. For this analysis, $\sigma$ was estimated on the following basis. For run 17, in the region of a flow rate of 13 GPM, the raw data showed that the pressure at station 3 remained constant, while the pressures at stations one, two, and four were changing.

Thus some mechanism was acting to maintain the pressure constant at station three. The most reasonable hypothesis for this mechanism is cavitation at station three. On this basis the critical cavitation parameter, $\sigma_c$, becomes 0.086. The lowest cavitation number observed is 0.027 on Run 9 at station 3 at maximum flow. If the estimate for $\sigma_c$ is reasonable, then a value of 0.027 is well into the cavitation regime.

It should be pointed out that cavitation is probably occurring at other conditions (higher values of $\sigma$) but is almost certain in the cases referenced above.

Looking at the three dimensional aspects of the flow can shed some light on the frequency of cavitation. The pressure on the surface of the ball can be estimated in the following manner. Consider the geometry in Run 9 at station 3, as shown in Figure 2-19. The flow is bounded on the outside by a $30^{o}$ divergent cone and on the inside by the ball with a radius of 0.187 inch. If the flow is full, the streamlines should change continuously between the boundaries. The streamlines near the outer wall will be straight; those near the ball curved. Where the flow is curved, a pressure gradient normal to the flow streamline must exist. This is written

$$dP/dy = -\rho a$$

where P is the pressure, y the direction normal to the streamline, $\rho$ is the fluid density, and a is the fluid acceleration. If we know the velocity on the streamline the acceleration is given by

$a = \frac{V^2}{R}$ where R is instantaneous radius, and V is the fluid velocity.

We now have

$$dP/dy = -\frac{\rho V^2}{R}$$

Assume $\frac{1}{R}$ is 0 ($R \rightarrow \infty$) at the outer wall and increases linearly to the curvature of the ball ($\frac{1}{R}$ directly proportional to y). A further assumption will be made that V is the mean velocity of the flow. Rearranging and substitution gives

$$dP = -\frac{\rho V^2 (144)}{(0.187)(0.0302)} y \, dy$$

Here 144 is the conversion factor between $in^2$ and $ft^2$, 0.187 is the radius of the ball, and 0.0302 is the gap dimension $\Delta y$. Integrating this along y, with y varying from the tapered shoulder to the surface of the ball gives (assuming that V can be taken to be independent of P):

$$\Delta P = - \frac{V^2}{2g} \frac{(.0302)}{(.187)} = - 0.156\ V^2\ (psf)$$

where g is the acceleration due to gravity and the units (psf) are pounds per square foot. Applying the last equation to the data (which requires adding 14.7 psi to the measured wall pressure) predicts that the calculated absolute pressure on the surface of the ball is negative for a flow rate of 8 GPM. A negative pressure indicates the existence of cavitation in the flow. Therefore, it is likely that cavitation began at a somewhat lower flow rate and became controlling at 10 GPM.

In actual regenerative fixtures, cavitation is highly unlikely to occur because of the much higher back pressure ($\sigma \simeq .8$). Therefore, none of the data obtained where cavitation is suspected is applicable to the firing conditions. Flow separation, however, is still likely to occur under the conditions to be found in a regenerative fixture, provided that the geometry of the flow passage is sufficiently complex. If the flow is full and bounded (completely surrounded by liquid and metal), the separation bubble will contain recirculating liquid and mixing losses will be high, thus reducing the net mean flow rate. If the bubble is ventilated by gas, the losses may be less. However, if the gas is hot, ignition and flame holding are likely to occur.

## Section 3

## TESTS WITH THE STEEL ORIFICE TESTER

### 3.1 BACKGROUND

In the cold flow tests an inert fluid (water) was pumped through the ball check injector with only atmospheric back pressure. Although the flow exhibited separation and cavitation, relatively high discharge coefficients were obtained (up to .8).* This was strikingly different from the values calculated from 25mm gun fixture firings with this injector orifice, which were typically about 0.3.** The gun firings also showed a disturbingly high frequency of piston reversals, on the order of 25 to 30%.

As already mentioned, the flow conditions which can be generated in the plastic fixture are several orders removed from the conditions that occur in an actual regenerative gun test fixture. A special steel fixture was therefore designed and fabricated for the purpose of better understanding the nature of the flow of a monopropellant under regenerative fixture conditions. A description of this fixture may be found in the Introduction. A layout drawing of this test device is shown in Figure 1-2. This fixture is referred to as the orifice tester. In particular, the layout drawing indicates the position of the various sensors. Combustion chamber pressure is measured at P9 and P10. The pressures at two points in the flow in the injector are measured at P1 and P2. The pressure at the entry to the orifice is measured at P3. Pressure is also measured at two points in the propellant reservoir, P4 and P5. In addition, light sensor taps are located in the same plane as P1 through P5, but are at a 90° angle with respect to the pressure transducers. The light sensors are designated L1 through L5.

The pressure gauges used were PCB model no. M0119A, although equivalent Kistler gauges were used in additional tests performed at the General Electric Armament Systems Department, Burlington, Vermont (GEA & ESD). The light sensors were of the phototransistor type, with maximum sensitivity in the near infrared. A more detailed description of the light sensors may be found in the "Liquid Propellant Technology Annual Report," February, 1980.[1] The motion of the hydraulic multiplier piston was tracked using a linear optical encoder of special design. The piston differential area ratio was 1.27.

*For these tests, the ball was free floating against the spring in the spline.

**Some tests with the fixed check valve also had low $C_D$. See Run Nos, 9, 14, and 15.

This was a difficult experiment to instrument. The problem lay with the size of the pressure taps and their length. As can be seen from the drawing, the sensing hole penetrates through the carrier holding the injector. This additional length can act as a filter reducing the frequency response of the sensor. Far worse, however, is the disturbance that the presence of the sensing hole has on the flow itself. The standard sensing port diameter is 0.062 inches, which is on the order of the characteristic dimension of the spline. An attempt was made to mitigate this problem by reducing the sensing port dimension to 0.03 inches and packing the sensing hole with grease for every firing. However, this is not a very satisfactory solution. It should be remembered in the succeeding discussions that the pressures measured in the flow passage include a component of unknown magnitude which represents the interaction of the flow with the sensing port. This problem was not as severe with the light sensors, as these taps were filled with a fused quartz rod.

## 3.2 TEST DESCRIPTION

Four series of tests were run: (1) ball check injector; (2) straight orifices, filled with grease; (3) straight orifice filled with nylon screws plus 1.3 $cm^3$ of unintentional air; and (4) straight orifices filled with nylon screws and with the air purged. Conditions (2), (3), and (4) are illustrated in Figure 3-1. All of these tests were performed with NOS-365 propellant. The test firings, together with relevant parameters and results, are listed in Table 3-1. (The first test was performed at the Princeton Combustion Research Laboratory and is not listed in this Table. See Note below.)

During the first test series, increasing wear of the vents connecting the primer cavity with the combustion chamber resulted in a general trend to increasing rates of pressure rise in the chamber. This increasing pressure rise rate confirmed the need to design the system in such a way that the propellant reservoir pressure can quickly adjust to changes in the combustion chamber pressure.

## 3.3 FIRINGS WITH THE BALL CHECK INJECTOR

The cold flow tests with the ball check injector yielded a discharge coefficient approximately 2-3 times higher than the value calculated in 25-mm firings using this injector. The first three† tests, however, in the orifice tester yielded a discharge coefficient very close to that inferred from the 25-mm fixture firings. The fourth test had the lowest rate of pressure rise of the series and a very different result. The pressure data from this test are shown in Figure 3-2. A small amplitude, 7 kHz oscillation is present in P1, P2, and P3. The inferred discharge coefficient was relatively large, within 30% of the value determined in earlier flow tests (about twice the previous value).

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†Note: The results from the first test are reported in the "Liquid Propellant Technology Annual Report," February, 1980 (Reference 1).

TABLE 3-1. SUMMARY OF TESTS PERFORMED WITH THE STEEL ORIFICE TESTER

| Run # | Propellant Volume ($cm^3$) | Peak Chamber Pressure (kpsi) | dPc/dt (kpsi/msec) | Injector Type | Results/Comments |
|---|---|---|---|---|---|
| 310:11 | 10 | 16.0 | 3.5 | Ball Check | Piston full travel, $P_1$ low, Low $C_D$, Burn in Orifice |
| 311:09 | 24 | 26.6 | 3.3 | Ball Check | Piston Full travel, $P_2$ Low, Low $C_D$, Burn in Orifice |
| 316:10:19:20 | 51 | 15 | .75 | Ball Check | Piston full travel, $P_{10}$ Low, High $C_D$, No Burn in Orifice |
| 316:14:43:31 | 52 | 30 | 4.0 | Ball Check | Fire in Prop. (Begun in Orifice) |
| 334:10:42:34 | 10 | 22 | 6.0 | Straight through holes, grease packed | Reversed |
| 334:15:03:51 | 30 | 30 | 6.4 | Straight through holes, grease packed | LS-3 turns on after plateau pressure is reached. No piston position signal. Reversed |
| 347:11:26:02 | 23 | 23 | 6.5 | Straight through holes, grease packed | Reversed |
| 351:13 | 23 | 23 | 5.6 | Nylon Screw + 1.3 cc's of air | LS-3 turn on after plateau pressure is reached. Reversed |
| 353:15:04:21 | 24 | 24 | 7.3 | Nylon Screw + 1.3 cc's of air | LS4, LS3, LS2, LS1 turn on in sequence Reversed |
| | | | TESTS PERFORMED AT GEA & ESD | | |
| OT-3 | 25 | 38.0 | 2.7 | Nylon Screw Without Air | Reversed |
| OT-4 | 25 | 36.0 | 2.0 | Nylon Screw Without Air | Reversed, $P_3$ late in showing pressure rise. |
| OT-5 | | 31 | 1.0 | Nylon Screw Without Air | Full travel. |
| OT-6 | 25 | 36 | 1.2 | Nylon Screw Without Air | Reversed. LS2 turns on after pressure peaks. |
| OT-7 | 25 | 44 | 1.0 | Nylon Screw Without Air | Reversed. |

Note: In addition, one test with the ball check injector was performed at the PCRL.

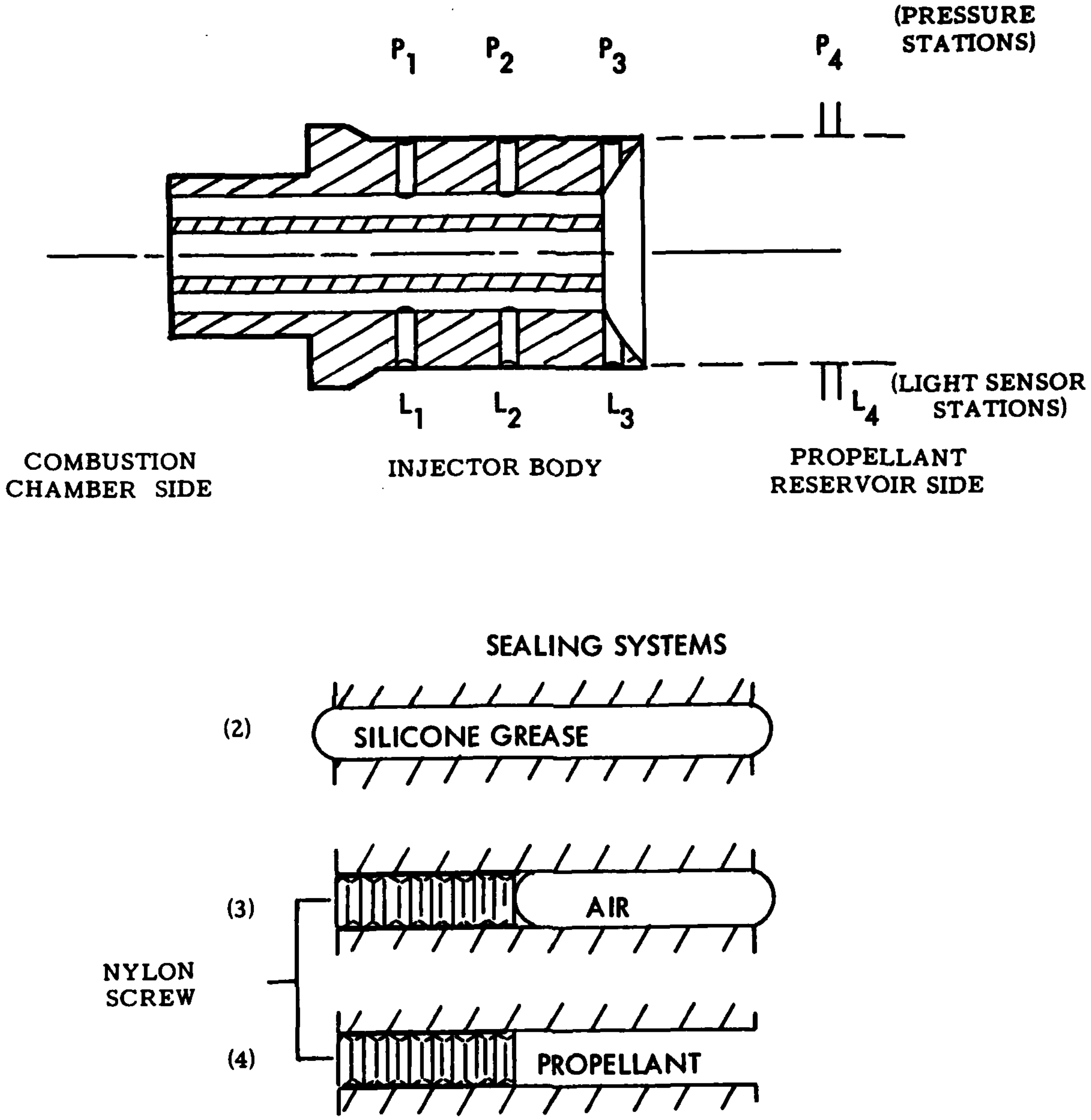


Figure 3-1. Schematic of the Configuration of the Straight Through Holes Tested.

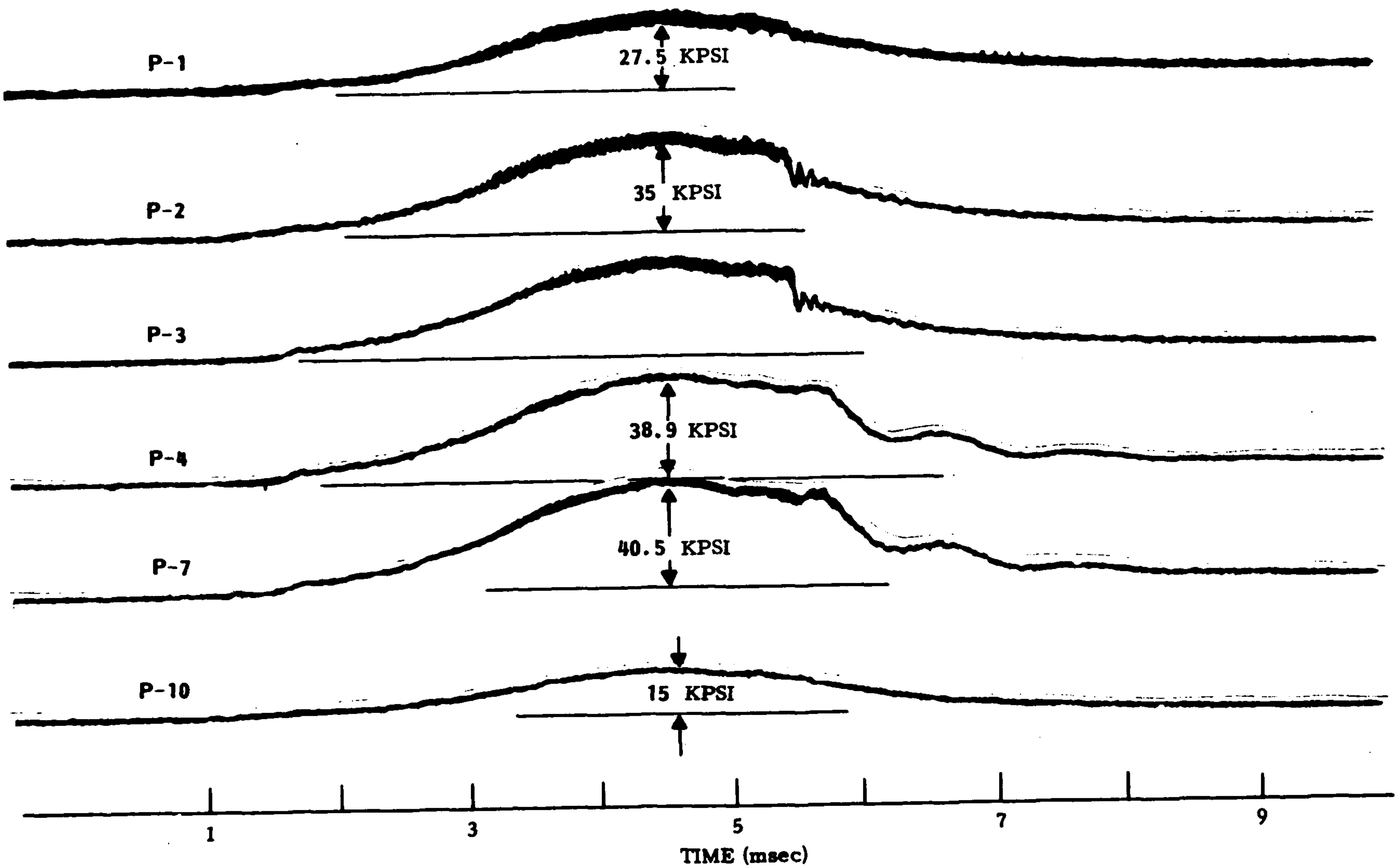


Figure 3-2. Pressure Data from Test 316:10:19:20 (Hand Traced).

This firing was repeated the same day, but the initial pressure rise rate was much higher and a low $C_D$ was inferred. The pressure data from this firing are shown in Figure 3-3. At about 3 msec into the firing oscillations were observed at station P3 (upstream of the ball). These oscillations became more severe and the piston reversed.

In summary, of the five tests performed with the ball check injector, only one exhibited a discharge coefficient comparable to that measured in the earlier flow tests in the plastic fixture. Of the four tests which exhibited a low discharge coefficient, two had piston reversals. An hypothesis, therefore, is that for tests with a low discharge coefficient, propellant burns in the injector orifice itself.

The discharge coefficient, $C_D$, is defined by the relationship

$$V = C_D \sqrt{\frac{2g\Delta P}{\rho}}$$

where V is the flow velocity, $\Delta P$ is the pressure drop across the orifice, $\rho$ is the liquid density, and g is the acceleration of gravity. P is measured directly and V is calculated from the injection piston velocity (including the correction for the compressibility of the column). This allows calculation of $C_D$. In particular,

$$C_D \propto \sqrt{\rho}$$

Thus, if the orifice were filled with gas rather than liquid, the density would be over estimated by a factor of about 5, resulting in an underestimate of $C_D$ by about 2.2. This roughly agrees with the results measured from the data.

The calculation above is completely correct only for single phase flow. Here it is assumed that liquid enters the flow passage but that gas leaves it. Hence the correct value for $C_D$ is likely between the $C_D$ calculated assuming pure liquid flow and the $C_D$ calculated assuming pure gas flow. For the conditions of the tests, though, it would be expected to be closer to the latter than the former.

In this vein it would appear that in the first three ball check injector firings the propellant was ignited as it came around the ball and was sprayed into the hot primer gases that filled the injector downstream of the ball. Combustion persisted as in a stable flame holder. In the fourth test, the slower pressurization rate (and consequently the enhanced ability of the propellant reservoir to adjust to the rising combustion chamber pressure) permitted the injected liquid to flush out the gas in the injector so that the injector flow was pure liquid.

The last shot of this group had the highest rate of pressure rise of this group. Ignition again occurred as the propellant was sprayed around the ball. In this case, the combustion was so violent that it was able to disrupt the flow upstream of the ball. This coupling resulted in the pressure oscillations that grow in magnitude up to approximately 6 ms where the flame comes completely out of the orifice and into the reservoir resulting in the observed reversal.

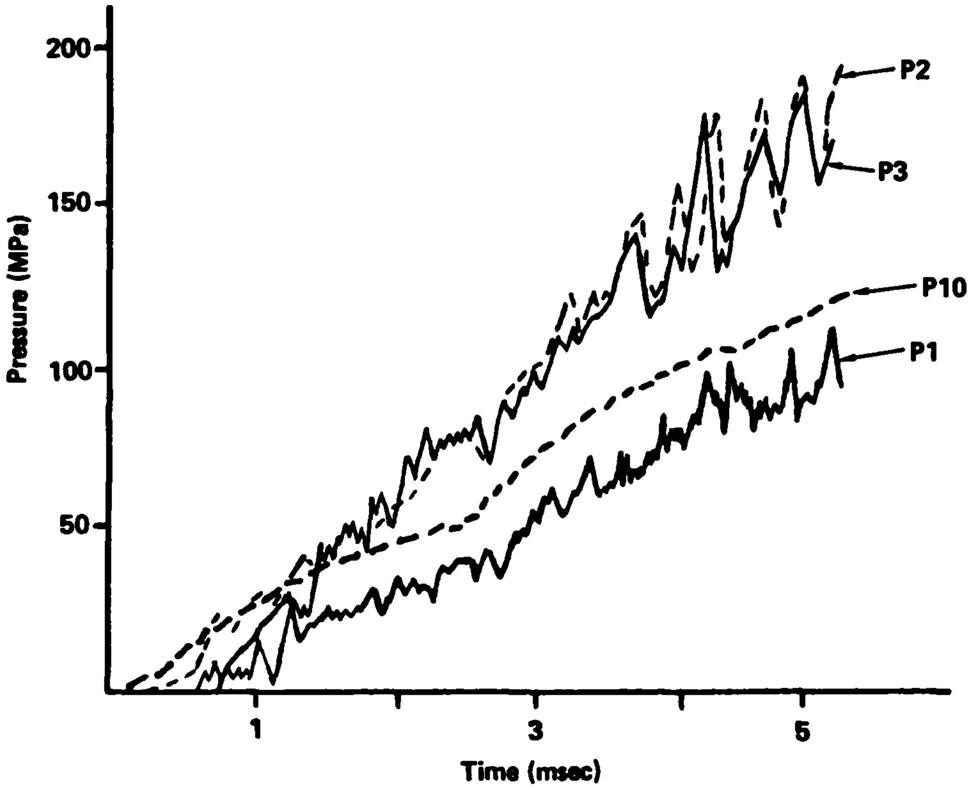


Figure 3-3. Pressure Data from Test 316:14:43:31 (Hand Traced).

### 3.4 TESTS WITH STRAIGHT THROUGH HOLES

#### 3.4.1 GREASE PACKED

The remaining tests were performed in the injector configuration that replaced the ball check injector. It is shown schematically in Figure 3-1. It consists of seven 0.109 inch diameter, 2 inch long holes, six on the circle and one in the middle. This orifice configuration was chosen for further study because, unlike the ball check injector, it had performed very well in the original regenerative fixture firings. Since these orifices are open, they must be plugged before firing to prevent leakage prior to ignition. For this purpose, the injector holes were packed with DC-111 grease. It should be noted that, in this configuration, the pressure taps sense the pressure in only one of the seven orifices. In the same manner, the light sensors sense conditions in a different flow passage. No direct information is available about the flow conditions in the other five passages.

Three tests were performed in this configuration. In all three the hydraulic multiplier piston reversed, indicating combustion in the propellant reservoir. The ignition source could have been adiabatic compression of entrapped ullage, but it is more likely to have been blowback of hot combustion chamber gases into the propellant reservoir.

The pressures for the low pressure startup region for these three tests are shown in Figure 3-4. The initial rate of pressure rise indicates that if the grease in all seven holes regressed uniformly*, the propellant reservoir pressure would rise sufficiently to stop this retreat before all the grease was pushed into the reservoir. However, since the holes were hand packed, hole to hole variations in the amount of retreat would be expected. The hole with the weakest grease barrier would permit gas ingestion sooner.

#### 3.4.2 PACKED WITH NYLON SCREWS (WITH AIR ENTRAPMENT)

Since the ignitions observed in the propellant were thought to be due to gas tunneling through the grease, a solid barrier was introduced in the next series of tests. One inch long nylon screws, with heads removed, were used. These were lightly greased and pushed in from the combustion chamber side. They were less than half the length of the orifice and in retrospect, the loading procedure allowed approximately 1.3 cc's of air to remain in the flow passage adjacent to the propellant reservoir. This is illustrated in Figure 3-1.

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*This calculation invokes only the equation of continuity. Suppose that V is the propellant reservoir volume and that it is necessary to raise its pressure by a small amount $\Delta P$ by pushing on the grease in the injectors. The volume V must be reduced by $\Delta V = \frac{\Delta P}{B} V$, where B is the propellent compressibility.

The volume swept out by the grease in the i-th hole is given by $b_i = Ax_i$, where A is the hole area and $x_i$ is the displacement of the grease in the i-th hole. The sum of all $b_i$ must be equal to $\Delta V$. The smaller the $x_i$ for one hole, the larger must be the $x_i$ for another hole.

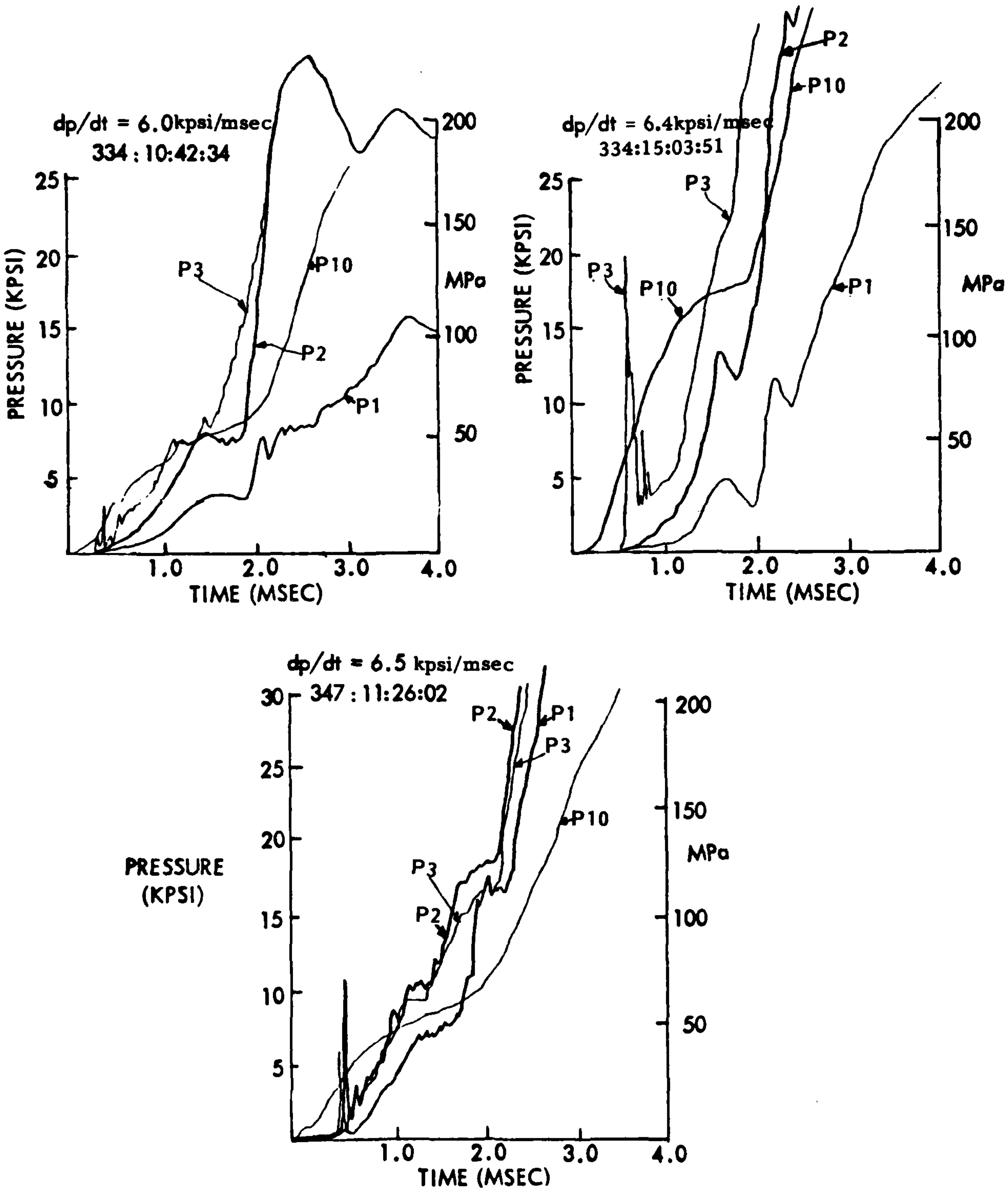


**Figure 3-4. Pressure Data for Startup Region for Tests 334:10:42:34, 334:15:03:57, and 347:11:26:02 (Hand Traced).**

sealing system (3). Two tests were fired, both of which promptly reversed. The last shot had a very high pressurization rate (7.3 kpsi/msec; the upward trend in pressurization rate during these tests was due to erosion of the booster vents). The pressure data from this firing are shown in Figure 3-5. On this shot the pressure and light sensor data indicate that major combustion began deep in the propellant reservoir. Whether this was due to adiabatic compression or gas blowback cannot be unambiguously determined, but it is clear that this was far too hard a start for such an hydraulically soft system, with 19 inches of fluid to compress (propellant plus water).

### 3.4.3 PACKED WITH NYLON SCREWS (WITHOUT AIR ENTRAPMENT)

With the lessons learned from the previous tests, a new series of tests were performed at GEA & ESD. The fill procedure was modified to eliminate the entrapped air. The replaceable booster vents were inspected and replaced as necessary to maintain the desired pressurization rate of about 2 kpsi/ms. Both of the first two firing (tests OT-3 and OT-4 (Table 3-1)) reversed with the first activity showing up in the orifice (station P2). The pressure data from these two tests are shown in Figures 3-6 and 3-7 respectively.

The nominal pressurization rate was then reduced to 1 kpsi/ms and a "classic" firing resulted. The pressure data from this firing are shown in Figure 3-8. Note the smooth development of all the pressures until completion of injection at about 10 msec. When this shot was repeated (see Figure 3-9 for the pressure data), a reversal resulted. The pressure in the propellant (station P3) showed a ringing that suggested the presence of air. It was hypothesized that this could have been air trapped in the "O" ring grooves of the separator piston. The last test was fired with a new separator piston without "O" rings. A reversal resulted none the less. The pressure data from this last firing are shown in Figure 3-10.

A plot of the calculated separator piston motion for tests OT-3 to OT-7 is shown in Figure 3-11. The calculation is based on the measured piston motion and the measured liquid pressure. This latter is used to determine what fraction of the measured piston motion went simply into compressing the fluid column. This correction factor was applied to both the propellant and water columns. As can be seen in this figure, only in test OT-5 did the separator piston achieve full travel.

Even though all five of these tests look different, they all, with the exception of OT-5, the "classic" result, show activity in the orifice prior to any major manifestations of combustion. This is similar to the ball check firings, where flame holding is believed to have occurred. The only obvious flame holder in this orifice is the instrumentation port (1/3 the dia. of the hole). However, a completely satisfying explanation for these results is not available. A brief discussion of flame holding in a straight orifice is presented in Appendix A.

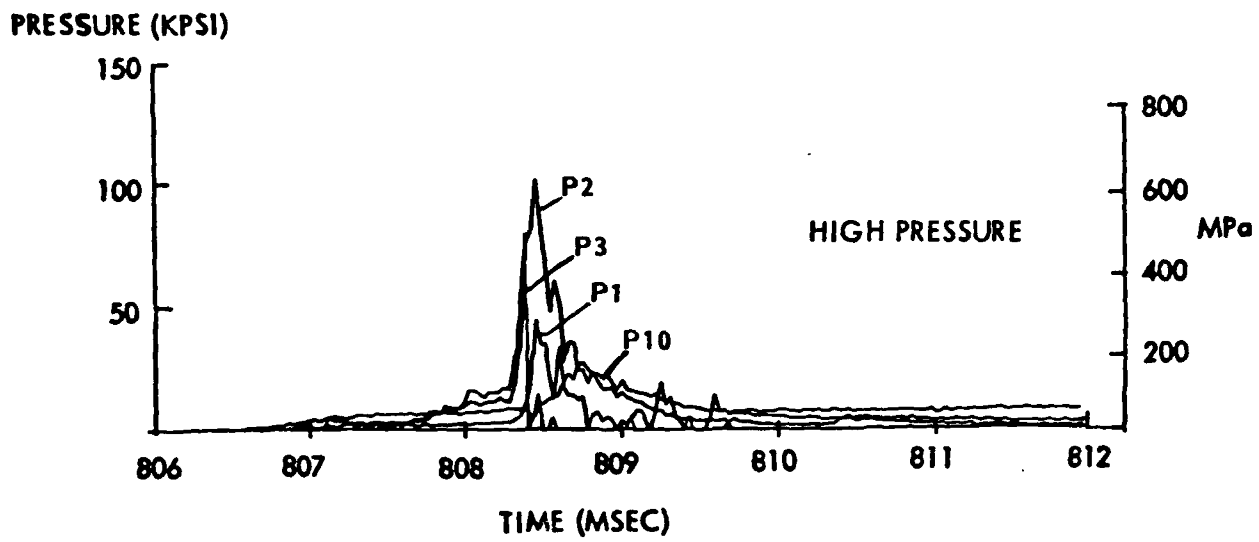


Figure 3-5. Pressure Data from Test 353:15:04:21, Nylon Screw with Air Entrapment (Hand Traced).

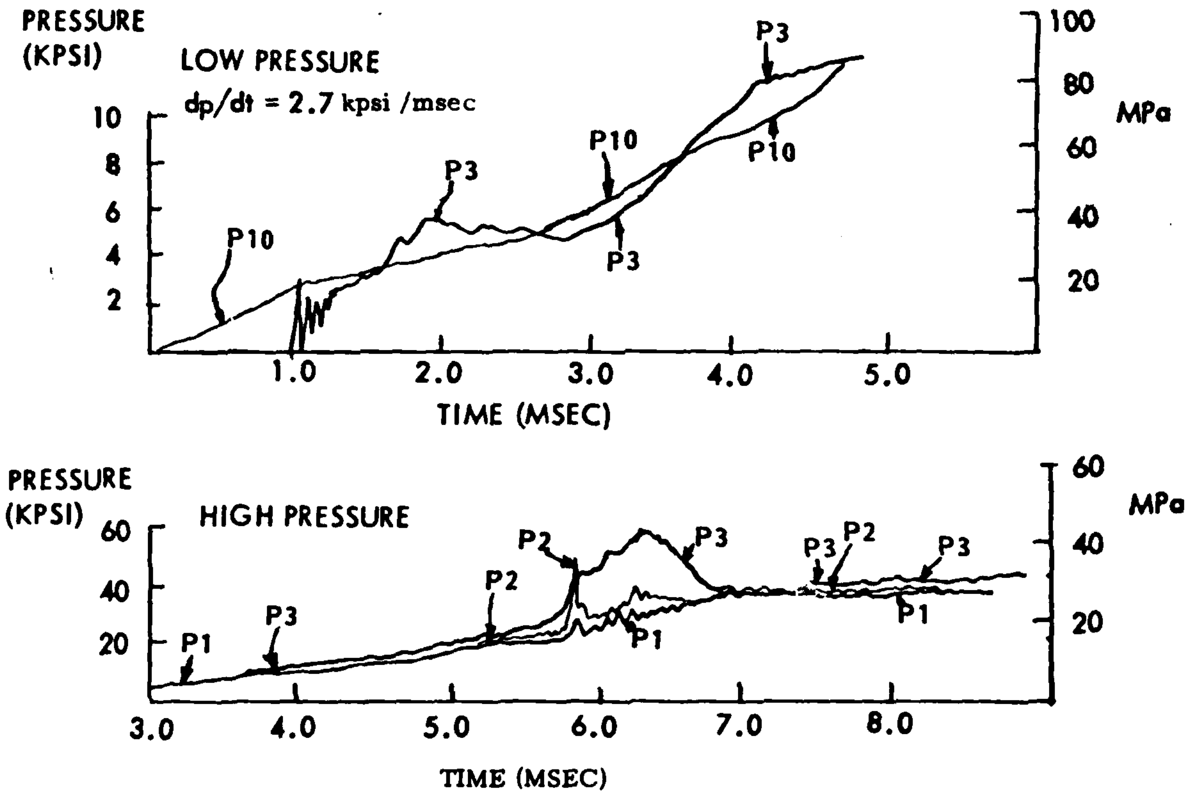


Figure 3-6. Pressure Data from Test 0T-3 (Hand Traced).

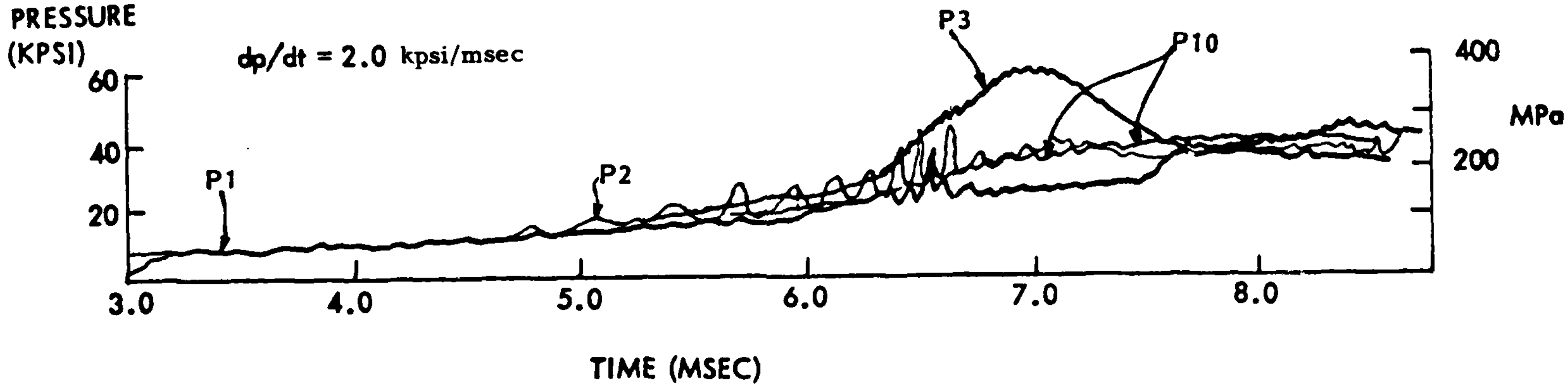


Figure 3-7. Pressure Data from Test 0T-4 (Hand Traced).

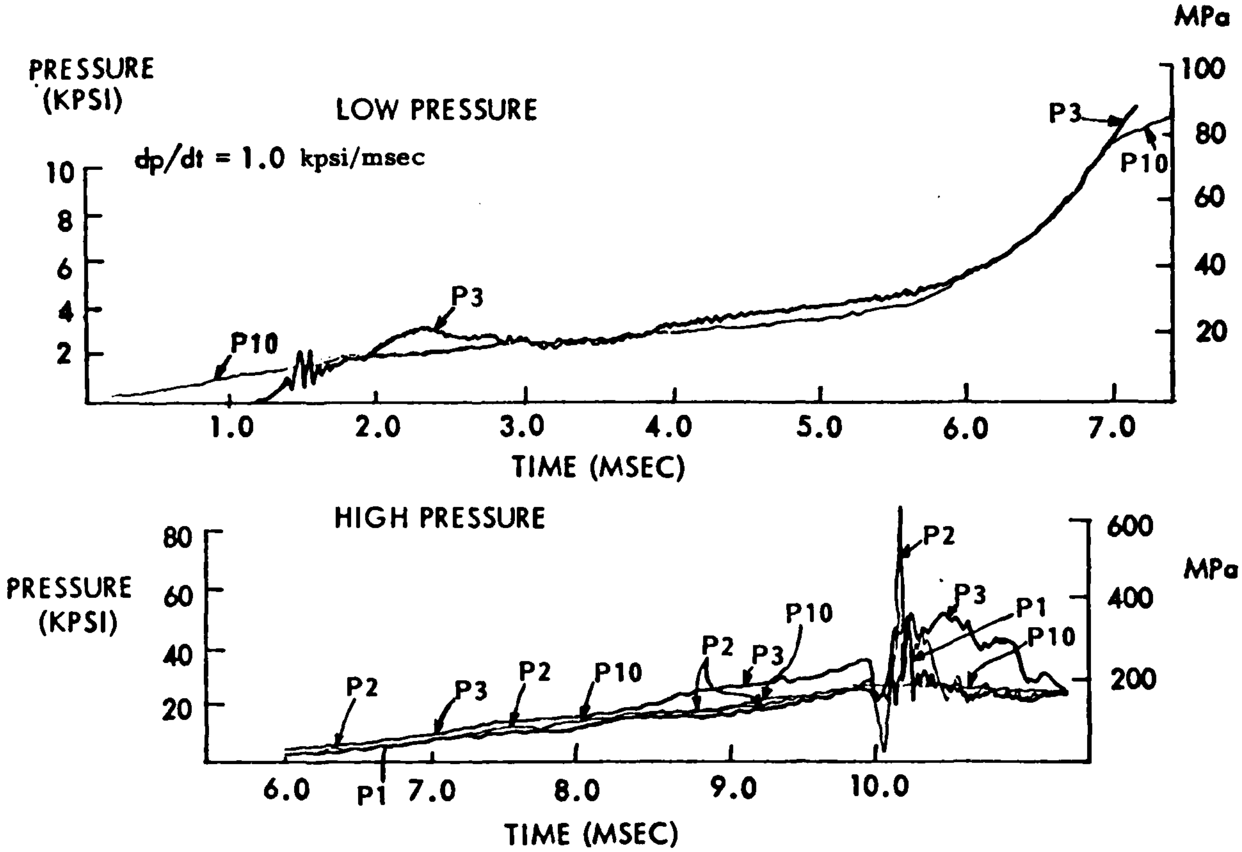


Figure 3-8. Pressure Data from Test 0T-5 (Hand Traced).

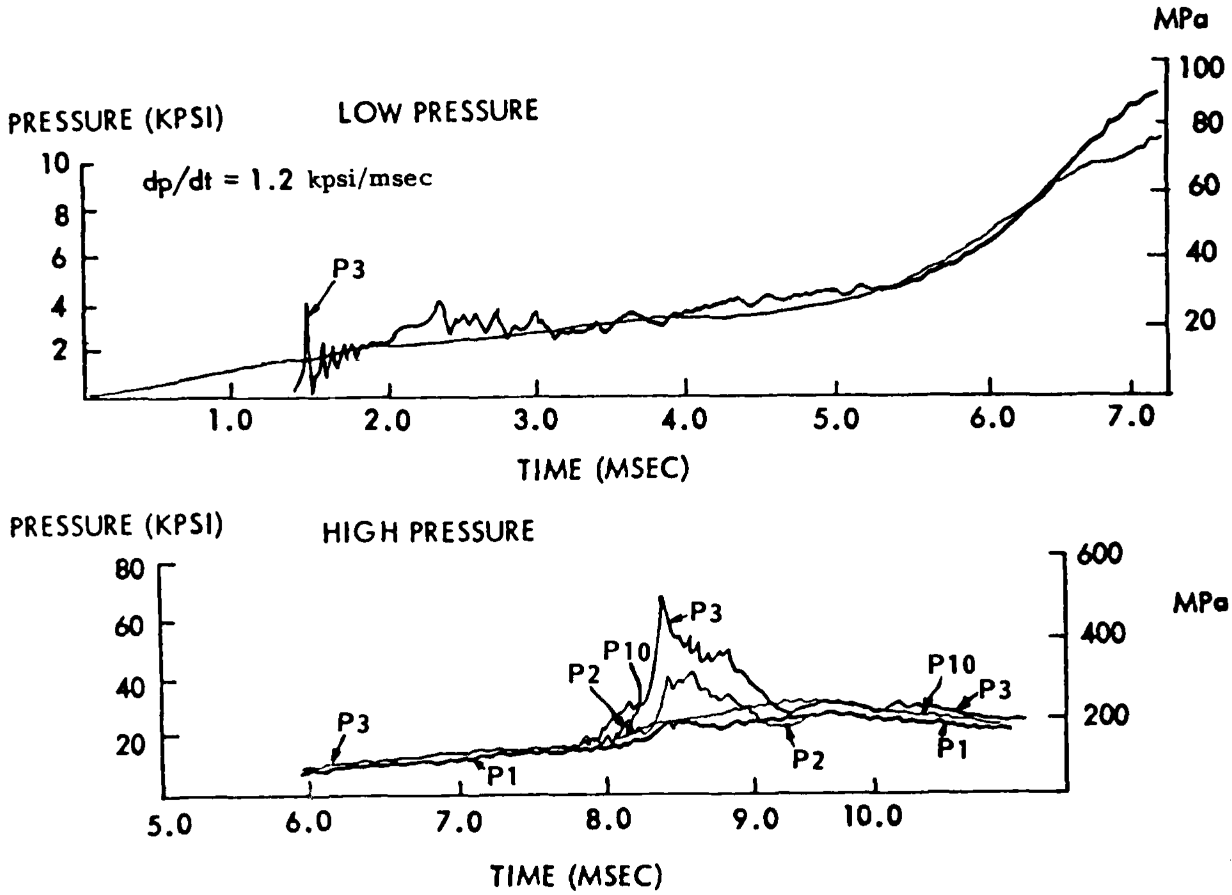


Figure 3-9. Pressure Data from Test 0T-6 (Hand Traced).

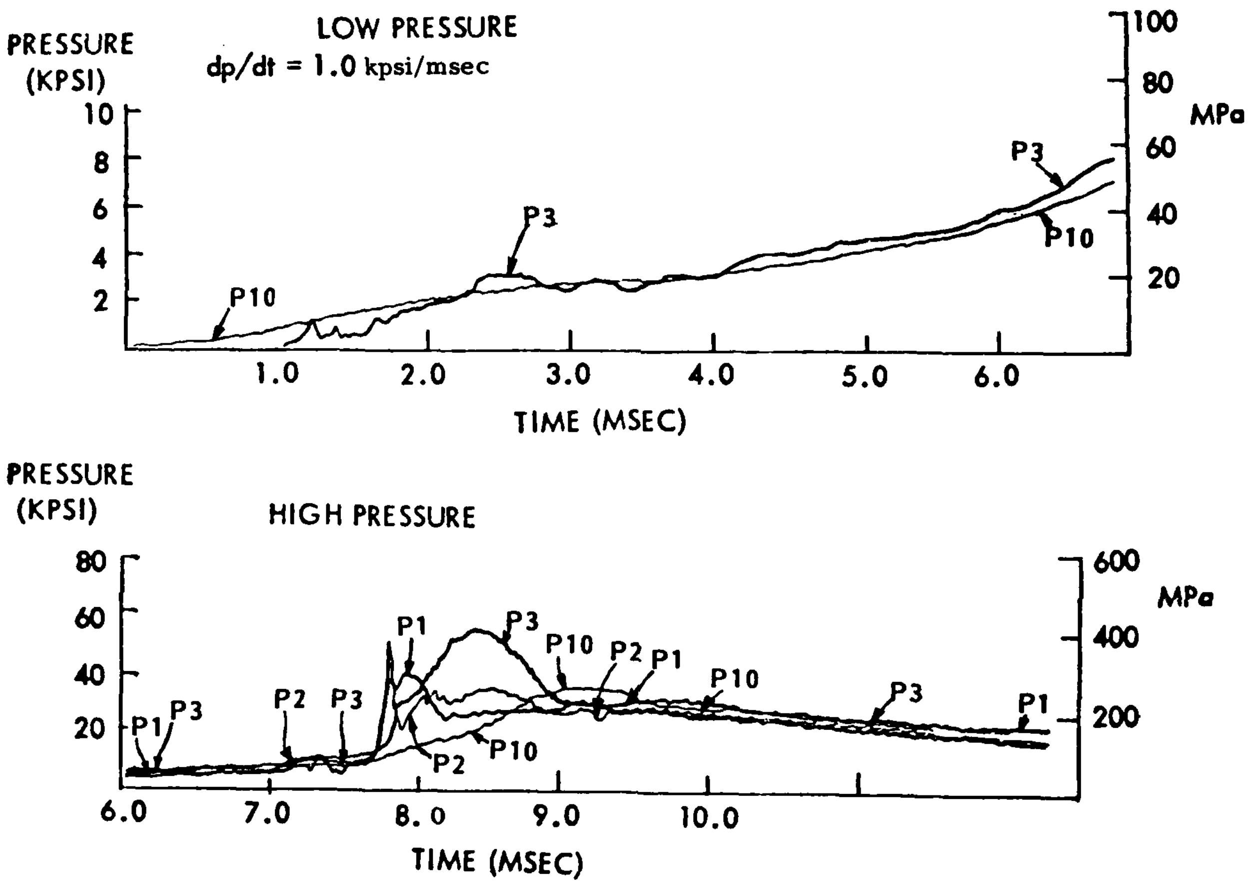


Figure 3-10. Pressure Data from Test 0T-7 (Hand Traced).

## Section 4

## DISCUSSION, CONCLUSIONS, AND LESSONS LEARNED

### 4.1 DISCUSSION AND CONCLUSIONS

The dominant characteristic of the test firings was the very high incidence of piston reversals, indicating combustion in the propellant reservoir. Of the 15 tests performed, 9 resulted in piston reversals, and in only two does it appear that there was no evidence of combustion occurring in the injectors. This is a rate of incidence far higher than ever observed even with the worst performing regenerative firing fixture. For the latter case the incidence of piston reversals was only 20 to 30%. Even in the tests of Concept V, the incidence of piston reversals associated with blowback of hot gases was only one in thirty. In Concept VI, no reversals have been observed in over 60 tests.

The basic goal of this effort, to study the flow of a monopropellant through an injector under the conditions to be found in a regenerative firing fixture, was not met. This is very disappointing, as it may mean that such flows cannot be studied directly, but rather that their nature must be inferred within the context of a normal regenerative firing fixture.

It was known beforehand that this experimental approach would have to overcome two difficult problems: ensuring that the propellant reservoir pressure properly follows the rise in combustion pressure; and that the instrumentation not interact too strongly with the flow. It may be that neither of these requirements was achieved.

It must be pointed out that the problem is more likely associated with hot gas generated or entrapped in the injectors rather than because of ullage compression ignition. This is based on two arguments. First, a considerably larger number of test firings had been performed earlier in 25-mm and with multiple holes through the injection piston face. The propellant used was also NOS-365. The fill procedure used was essentially the same as that used in loading the orifice tester, and hence should have been subject to the entrapment of a comparable amount of ullage. However, the observed frequency of piston reversals was far less than that observed in the orifice tester. Second, the test with NOS-365 performed at PCRL[2], under conditions as least as severe as the conditions used in the present study, did not result in ignition from adiabatic compression.

The weakness in the first argument is that the orifice test fixture and the 25-mm fixture have significantly different cycle times. In the orifice tester, high pressure is maintained for times in excess of 10 msec while in the 25-mm fixture, high pressure is maintained for less than half this duration.

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2 Messina, N.A., Ingram, L.S., Camp, P.E. and Summerfield, M., "Compression - Ignition Sensitivity Studies of Liquid Monopropellant in a Dynamic-Loading Environment," Princeton Combustion Research Laboratories, Inc., Princeton, New Jersey 08540

It can be legitimately argued that the cycle time associated with the 25-mm fixture is simply too short to generate sufficient gas buildup. Indeed, long delays, on the order of 10 to 20 msec, have been seen in association with compression ignition.[3] However, compression ignition tests performed by the PCRL[2] with NOS-365 indicate that the pressure rise rates used in the present test series should have been low enough to prevent compression ignition. However, there is uncertainty associated with this last statement in that the ullage character in the two cases was quite different. In the PCRL tests, it was dispersed throughout the propellant volume as small bubbles, typically 0.001 inches in diameter. In the orifice tester, whatever ullage was present would have been in the form of a few large single bubbles.

The above arguments do not indicate that compression ignition never occurred, but rather that it was unlikely to be responsible for the majority of the observed reversals. It should also be remembered that, in at least three tests, a rather large ullage was probably present in the injectors themselves.

This leaves the injectors as the most likely ignition source, either through blowback of hot combustion gases, or perhaps due to an interaction of the flow with the sensor taps, or both. The first explanation is a very likely candidate, as the system which the injection piston must compress is relatively soft. The length of compressible fluid is nineteen inches. Since the majority of this liquid is water, its bulk modulus is about 300,000 psi. This means that, to achieve a steady state pressure of 30,000 psi in the propellant reservoir, the injection piston must move almost two inches. Any motion to compensate for propellant outflow through the injector must be in addition to this. In practice this means that a steady state is never achieved. Even in the last tests, with an initial combustion chamber pressurization rate of only 1 kpsi/msec, the piston motion data indicated that the piston was always accelerating. There may simply never have been as great a differential pressure across the injector as originally intended.

As indicated in the previous section, the low inferred discharge coefficient may be a strong indication that combustion is occurring within the injectors. The mechanism as to how this would occur is not entirely clear in the case of the straight through holes. Conceivably, the pressure taps may in some manner be involved. If, in addition, the differential pressure across the injector is lower than planned for, blowback of hot gases into the propellant reservoir could become likely. It is certain that they would initially be partly filled with hot gas because, during startup, the pressure in the combustion chamber is significantly higher than in the propellant reservoir.

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[3] J. Mandzy and K. Schaefer, General Electric Ordnance Systems, and J. Knapton and W. Morrison, BRL, "Progress Report on Compression Ignition of NOS-365 Under Rapid Fill Conditions," 17th (1980) JANNAF Combustion Meeting, <u>CPIA Publication 329, Vol. II, pp. 309-327</u>, Applied Physics Laboratory, Johns Hopkins University.

A further hindrance is that the pressure measurements in the injectors themselves (P1, P2, and to some extent P3) must be treated with suspicion. It is entirely too likely that much of the structure observed in these pressure traces is spurious, an artifice of the interaction of the flow with the pressure tap. To illustrate this one has only to observe the high speed movies made of the tests in the plastic fixture and observe the constant and rapid oscillation in position of the small air bubbles entrapped in the pressure taps.

Thus the basic concept for this test fixture, and the manner in which it was instrumented, appear to be unsuitable for studying the phenomenon of interest. The basic concept itself could be modified. The pressurization for the propellant could come from a source independent of the combustion chamber. Although more complicated, this approach is feasible. The question of noninteracting instrumentation is more difficult. At present, there appear to be no good solutions for measuring the pressure in the injectors without significantly disturbing the flow.

## 4.2 LESSONS LEARNED

Although the basic goals of this effort were not met, a number of useful lessons were learned. First, it was always believed that the system must be designed with mechanical parameters such that it can readily follow the normal expected rate of pressure rise in the combustion chamber. These tests confirmed the importance of this design principle and provided a number of graphic examples of the consequences when it was violated. These tests failed to yield quantitative criteria for the design of injectors, as had been the original hope. However, they did provide clues as to how they should be configured. They should be simple in shape with no sharp edges, turns, or sudden changes in flow area. The data may also indicate that excessively long injectors may be undesirable.

ACKNOWLEDGMENTS

The authors wish to express their thanks to Mr. K. Reever of GEOSD and Mr. D. Watt of GEA & ESD for conduct of the actual testing. The support, during testing, of Mr. S. Henderson and Mr. H. Hall of the Wright-Malta Corporation is appreciated. An especial note of thanks is due Mr. R. E. Mayer for his many contributions to the design of the hardware and the interpretation of the data.

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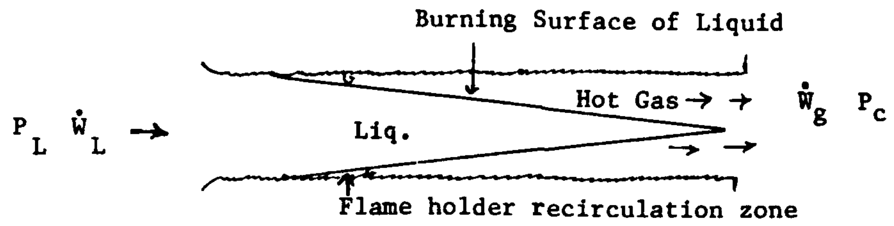
2. N.A. Messina, L.S. Ingram, P.E. Camp, and M. Summerfield, "Compression - Ignition Sensitivity Studies of Liquid Monopropellant in a Dynamic-Loading Environment," Princeton Combustion Research Laboratories, Inc., Princeton, New Jersey 08540, 1979.

3. J. Mandzy and K. Schaefer, General Electric Ordnance Systems, and J. Knapton and W. Morrison, BRL, "Progress Report on Compression Ignition of NOS-365 Under Rapid Fill Conditions," 17th (1980) JANNAF Combustion Meeting, CPIA Publication 329, Vol. II, pp. 309-327, Applied Physics Laboratory, Johns Hopkins University.

## APPENDIX A

## FLAME HOLDING IN A STRAIGHT ORIFICE

Consider the following scenario. Let $P_L$ and $P_C$ represent the propellant reservoir and combustion chamber pressures and $\dot{W}_L$ and $\dot{W}_g$ are the liquid mass flow rate entering the straight orifice and the gas mass flow rate leaving the orifice.




The question to be addressed is: Can flame holding occur in a straight orifice?

**FLAME HOLDING:**

Flame holding occurs when recirculation of hot gas continuously ignites fresh propellant. What is required is any flow passage where flow separation may occur (sudden increase in area, sharp turns, excessive roughness, etc.). The flow may be stable as long as hot gas is present and the liquid continues to separate.

If flame holding takes place, it will influence the flow by converting some (or all) of the propellant to hot gas. As the propellant reacts, the propellant/gas flow accelerates and the static pressure in the flow falls. Since the pressure at the end of the orifice cannot normally drop below the back pressure, this means that the mass flow rate through the orifice will be less when combustion takes place.

Let us look at an example from test OT-7 in the orifice tester. At 8.5 ms into the test, the reservoir pressure was 58 kpsi and the chamber pressure was 26 kpsi. For this differential pressure, the Reynolds number for the flow in the injector is about $3 \times 10^5$ assuming liquid flow. Assume a relative roughness[a] (E/D) of .002. This gives a friction factor[b] f of .024. The pressure drop $\Delta P_f$ through the orifice would be:

$$\Delta P_f = (f)\ (1/2\ \rho V^2)\ (L/D) \qquad L/D = 18.3$$

$$= .44Q \qquad (Q = \frac{1}{2}\rho V^2)$$

---

[a] Relative roughness is an hydraulic term that compares the magnitude of the average surface roughness E with the duct diameter D.

[b] The friction factor f is defined by

$$f \equiv \frac{\Delta P_f D}{QL}$$

where $\Delta P_f$ is the pressure drop due to friction, L is the duct length, Q is the dynamic pressure of the flow ($1/2\ \rho V^2$).

The estimated entrance loss is .3 Q (in hydraulics, a typical value for this entrance loss). The total driving $\Delta P_t$ must be:

$$\Delta P_t = .3\ Q + .44\ Q + 1.0\ Q = 1.74\ Q$$

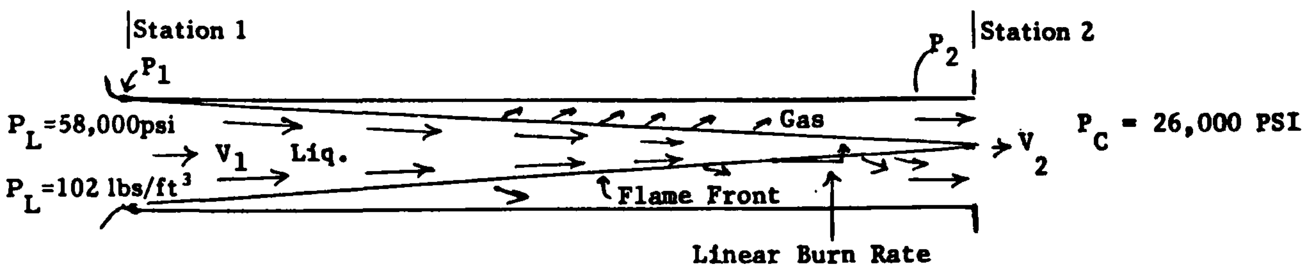
$$Q = \Delta P_t/1.74$$

$$1/2\rho V^2 = \Delta P_t/1.74 \qquad C_D = 0.758$$

$$V = 1500 \text{ ft/sec} .$$

For comparison, the linear burning rate of NOS-365 at 58 kpsi is about 0.8 ft/sec or about 2000 times slower than the flow rate.

Consider a case where the propellant is burned in the orifice and compare the required burning rate with the calculated value. We must calculate a new liquid flow rate since, due to the need to accelerate the gases generated, the combustion will use up a significant portion of the available $\Delta P$. The simplest case to handle is where a propellant enters the orifice and hot gas exits.




The liquid enters the orifice and accelerates to $V_1$ where the pressure has dropped to $P_1$. The propellant is ignited and burns radially in, consuming all the propellant before the exit. As the propellant is burned, the flow velocity goes up and the static pressure goes down. The pressure drop between the two stations is found from the momentum equation between the two constant area stations 1 and 2.

$$P_1 - P_2 = \rho_2 V_2^{\,2} - \rho_1 V_1^{\,2}$$

from continuity we also have

$$\rho_1 V_1 = \rho_2 V_2$$

we can find the pressure at station 1 from Bernoulli's equation

$$P_1 = P_L - \frac{1}{2} \rho_L V_1^{\,2} .$$

There are two approaches to solving this. The first assumes the exit flow is subsonic. This determines the exit pressure (this is simply the measured back pressure) and the exit velocity is then solved. When this is done in this case, the computed velocity is 4185 ft/sec. This is too high as it considerably exceeds sonic velocity and therefore contradicts the initial assumption. The second approach assumes sonic (supersonic is not possible in a constant area duct in steady flow) exit velocity. With the exit velocity fixed, the exit pressure is solved for assuming a sonic exit velocity of 3400 ft/sec. The exit pressure is found to be 37,200 psi. Since gas friction was not included, we will lower this to 30,000 psi. From continuity and the state equation, and by assuming an entrance density of 102 lbs/ft$^3$ and an exit gas density of 15 lbs/ft$^3$, we can solve for $V_1$.

$$V_1 = \frac{\rho_2}{\rho_1} V_2 .$$

$$V_1 = 500 \text{ ft/sec}$$

The residence time of the liquid in the orifice is found from the average velocity

$$\Delta T = \frac{L}{\bar{V}} = 8.5 \times 10^{-5} \text{ sec}$$

The burning rate required to consume the propellant (radial burn) is found from*

$$\dot{r} = \frac{\Delta R}{\Delta T} = 53 \text{ ft/sec}$$

This is over 50 times faster than the strand burning rate, but still much less than the liquid flow velocity assuming full flow. However, there is the possibility of burning rate augmentation (see Appendix B).

---

* $\Delta R$ is the radius of the flow passage. For all the propellants to be converted to gas before the end of the flow passage is reached, the flame front must cover this distance in the flow transit time $\Delta T$.

It has been shown that if flame holding takes place, it will affect the mass flow rate. If a significant portion of the propellant in the orifice lights off suddenly, the disruption can be more severe due to the water hammer effects. A simplified wave dynamic analysis of the orifice flow in test OT-7 between the times 7.8 msec and 8.0 msec is presented in Figure A-1. At t = 7.8 msec, assume a strong ignition occurs at station P1. Two effects happen. Because the pressure is so high , the flame front propagates both upstream and downstream. In addition, because of the strength of the explosion, shock waves are propagated in both directions and are reflected from the two ends of the flow passage. These events correlate with the pressures measured at the various gauge locations (see Figure 3-10). If the system response results in a reverse pressure gradient of sufficient duration, the hot gases in the orifice can flow into the reservoir and ignite the main charge. This is felt to be the cause of the observed reversals.

If the orifice surface is sufficiently rough, flame holding may be possible at any number of locations. This will permit "ratcheting". This occurs when brief flow reversals allow the combustion to latch onto a site upstream. This can lead to hot gas working its way upstream to the reservoir when the flow is sufficiently non-steady.

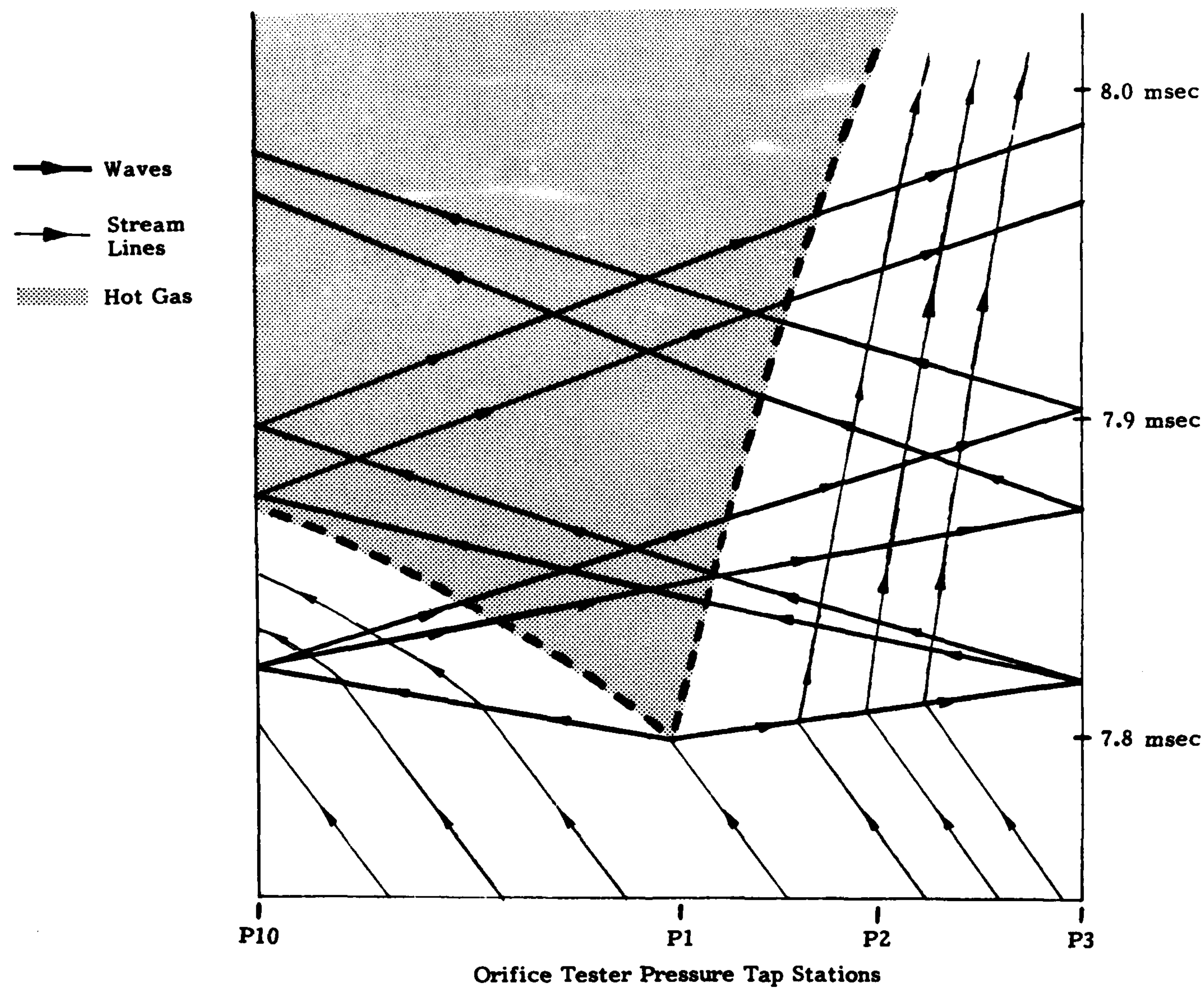



Figure A-1. Wave Diagram of Water Hammer Effect (Test 0T-7).

## APPENDIX B

## ESTIMATION OF POSSIBLE BURNING RATE AUGMENTATION IN THE ORIFICE

The flow in the orifice is highly turbulent ($Re_d \sim 1 \times 10^5$) . In turbulent flow the velocity vector of a particle fluctuates in magnitude and direction about the mean flow. The confinement of the boundary causes the normal component of the velocity to drop to zero near the wall. This causes the flow near the wall to be laminar (the laminar sub layer). When the flow separates, the turbulent core flow ruptures the unsupported laminar layer. The surface of a free jet is seen to be very rough as the turbulent eddies emerge from the surface. This rough surface will permit faster burning due to its higher surface area. We will try to estimate what this higher area is.

Very near the wall, the "Law of the Wall"[a] applies, giving us

$$Y^+ = U^+$$

$$Y^+ < 10$$

But $U^+ \equiv U/V^*$ , $Y^+ \equiv YV^*/\nu$ , $V^* \equiv \sqrt{\tau_w/\rho_w}$ and $\tau_w \equiv C_f \; e \; U^2_{av} / 8$

---

[a] The standard description for the velocity profile of a turbulent flow very very near the wall is given by

$$Y^+ = U^+ + e^{-kB} \left[ e^{kU+} - 1 - kU^+ - \frac{(kU^+)^2}{2} - \frac{(kU^+)^3}{6} \right]$$

In the laminar sublayer, defined by $Y^+ < 10$, this reduces to

$$Y^+ = U^+$$

Definitions of these terms, and further discussion of boundary layer theory can be found in numerous textbooks, cf. Frank M. White, "Viscous Fluid Flow," McGraw Hill, 1974.

where U is the mean velocity at location Y, $\rho_w$ is the fluid density at the wall, Y is the perpendicular distance from the wall, $\nu$ is the kinematic viscosity, $\tau_w$ is the shear at the wall, $U_{av}$ is the average flow velocity in the duct. $C_f$ is the Darcy friction factor (skin friction). The nomenclature of White is followed here.[a] The friction factor may be calculated from the power law of Blasius

$$C_f \simeq 0.3164 \ Re^{-1/4}$$

For the case at hand, $C_f \simeq 0.0182$. Working through these equations results in

$$U = C_f \ U^2{}_{av} \ Y/8\nu$$

At a flow velocity ($U_{av}$) of 500 ft/sec, a $C_f$ of .02 is found. The kinematic viscosity of the propellant is $4.92 \times 10^{-5}$ $ft^2/sec$. This gives us a velocity near the wall of:

$$U = 1.26 \times 10^7 \ y$$

where y is the perpendicular distance from the wall.

When the fluid leaves the wall (i.e., the flow separates), this linear shear can no longer exist. The closest motion to this that does not produce shear is rigid body rotation.

It is suggested that the flow switches from uniform shear to uniform rotation (on a small scale) on separating from the boundary. The size of the regions of uniform rotation is limited by centrifugal force. The pressure excited at the surface of the region must be supported by surface tension. The surface tension $\tau$ is defined by

$$\tau = PR$$

where P is the pressure and R is the characteristic dimension of the region affected by the uniform rotation. For this case, the centrifugal force equation becomes

$$P = \rho\,(U/Y)^2\,R^2/2 = \text{radial pressure difference}$$

$$U/Y = 1.26 \times 10^7 \text{ rad/sec}$$

$$\tau = 0.003664 \text{ lbs/ft}$$

Solving for R gives $R = 3.1 \times 10^{-5}$ inches

If all the propellant were formed into droplets $3 \times 10^{-6}$ ft in radius and in contact with hot gases, they would burn in

$$\Delta t = \frac{R}{\dot{r}} = \frac{3 \times 10^{-6}}{.8} = 3.75 \times 10^{-6} \text{ sec}$$

where $\dot{r}$ is the measured linear burn rate of NOS-365 (in ft/sec). Since from Appendix A we computed a residence time of only $8.5 \times 10^{-5}$ sec it could most certainly burn before exiting the orifice. Not all of the liquid would form drops this small, but clearly significant augmentation seems possible once the flow separates and flame holding establishes itself.

We still haven't established a criterion for separation and flame holding. It has long been recognized that surface features smaller than the boundary layer thickness will not effect the bulk flow. There are several measures of the thickness of the boundary layer. One of them, the momentum thickness, ($\theta$), represents the displacement of the bulk flow by the boundary layer. It is therefore suggested that separation may take place if a surface irregularity exists greater than the boundary layer displacement thickness. $\theta$ is found from Whites "Viscous Fluid Flow" as*

$$\frac{\theta}{X} \sim .046\ Re_x^{-1/5}$$

where $Re_x$ is the Reynolds no. based on $(X = 2) \approx 10^6$

$$\theta = 5.8 \times 10^{-3} \text{ inch}$$

---

*$Re_x$ is the Reynolds number based on the characteristic dimension X of a body immersed in a flow.

Since the diameter of the transducer port is 30 x $10^{-3}$ inches it could most certainly lead to separation. Flame holding can only occur when hot gas is present. This could occur during the reverse flow of hot gases in the orifice during the start-up. This gas could then be available to ignite the liquid propellant as it passes this point later in the cycle.

Since the initial writing of the analysis, a similar approach to a similar problem was found in a paper given at the 6th International Symposium on Jet Cutting Technology, April, 1982. The paper, "The Coherence of Expulsive Water Jets" by Edwards, Smith and Farmer of the U.K., develops a mechanism to explain the dispersion of the droplets created around a water jet in air. They use the Radial Velocity gradient to compute the spin on a free (square) drop and go on to compute the lift normal to its flight path and thus its motion. Their work gives added credence to our analysis.

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